JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS.

VOL. XII. AUGUST, 1900. No. 3.

The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY
(Under whose supervision this number is published).

Commander H. WEBSTER, U. S. Navy.
Lieutenant Commander A. B. WILLITS, U. S. N. Lieutenant R. S. GRIFFIN, U. S. N.
Lieutenant Commander F. H. BAILEY, U. S. N. Lieutenant B. C. BRYAN, U. S. N.

CORROSION OF BOILER TUBES IN THE UNITED STATES NAVY.

BY LIEUTENANT COMMANDER WALTER F. WORTHINGTON, U. S. NAVY.

For the past ten years the boiler tubes used in the Navy have been almost exclusively lap-welded iron or steel fire tubes in Scotch, low marine, or in locomotive marine boilers, carrying a pressure of from 125 to 160 pounds. In the great majority of cases these tubes have been from No. 12 to No. 13 B.W.G., and have required renewal after from two to four years' service, counting from the date of commission of the ship and not from the time the tubes were delivered to the builders. In two cases the tubes were of No. 9 and No. 8 gage and lasted five and seven years respectively. In the latter case the log shows that for the first four years much cylinder oil was used and got into the boilers, and that combustion chambers and furnace crowns bulged. No doubt the tubes were partially protected from rusting by a coating of oil. The boilers of about half of the

ships had knobbled, hammered, charcoal-iron tubes and the other half low-carbon Bessemer steel.*

Little information of value for determining the relative durability of iron and steel tubes can be obtained from the records of the ships for various reasons. Small ships do much more cruising than larger ones; they have only two boilers, one of which is in constant use for auxiliary purposes, whereas larger vessels have four or more, and some of the largest have auxiliary boilers, so that the main boilers are not used except under way. Then some boilers leak at the seams much more than others; some ships have reserve tanks for fresh water; and the service in which a ship is employed and the part of the world in which she cruises make it possible or otherwise to fill up with fresh water whenever required, and make a difference in the quality of the fresh water obtainable. Some ships are retubed a little sooner than absolutely necessary to prepare them for a foreign cruise. Others are prevented from retubing as soon as required by the nature of the service in which they are engaged. Then some commanding officers and chief engineers wish to maintain the boilers in a higher state of efficiency than do others, who are more mindful of keeping down the cost of maintenance of the ship. Some ships use large quantities of cylinder oil, which coats and preserves the tubes at the expense of the other parts. In some ships large quantities of salt water are used, having the effect of depositing scale on the tubes which prevents them from rusting, but wastes fuel and reduces the maximum power of the boilers, besides endangering the combustion-chamber sheets and furnace crowns. These and other considerations which will suggest themselves to members of this Society, show that if we wish to learn anything about the corrosion of boiler tubes we must look elsewhere than to the records.

The reasons usually assigned for the corrosion of boiler tubes are as follows:

1. Fatty acids from decomposition of animal or vegetable oils.

* Seamless tubes were first procured for such boilers in the Summer of 1898, and not enough have been used to affect the general results as stated.

2. Hydrochloric acid due to the decomposition of the $MgCl_2$ in sea water at a high temperature.

3. Galvanic action.

4. Use of salt water.

5. Presence of carbonic acid in water.

6. Presence of air in water.

1. *Fatty Acids.*—As far back as 1879 a circular order was issued in our Navy prohibiting the use of any but mineral oils in steam cylinders. Since that time ample mineral-oil tanks have been provided, mineral oil has been easy to procure in any port, and it is cheap; there has been no temptation to use any other, and it may safely be assumed that litttle or no animal or vegetable oil has been used and no tubes have suffered from this cause.

2. *Hydrochloric Acid.*—Direct experiments have shown that under certain conditions it is possible to decompose ($MgCl_2$) magnesium chloride, at a temperature of 212 degrees Fahrenheit, setting free (HCl) hydrochloric acid. When such action takes place it appears that the HCl is always immediately appropriated by some base other than the boiler metal. The writer has often searched for but never discovered any acidity by testing the water with litmus paper, nor met any other engineer who had found acid in our naval boilers. Lewis states that "chloride of iron is not found in the water of the boiler, which would be the case if any corrosion or pitting were due to the action of free hydrochloric acid." Then, again, our naval boilers are not under steam on an average of more than one-third of their time, and no HCl can be generated when the water is cold. The usual practice is to fill all boilers with fresh water at the start, and to use only about half of the boilers for steaming, the others serving as fresh-water tanks. In this way but a small part of the make-up feed is taken from the sea. We may conclude that little if any damage is done to our tubes by the decomposition of the $MgCl_2$ in sea water.

3. *Galvanic Action.*—Experiments quoted by Howe have shown that when steel plates which have not been freed from mill scale are connected galvanically with plates free from mill scale, an

electric current capable of being measured with a galvanometer is set up. The mill scale is found to be about as effective as copper in producing the current. Tube makers condemn the practice of pickling the tubes on the ground that the acid soaks into the metal and cannot be neutralized by the ordinary method of washing the tubes in lime water upon removal from the pickling bath.* There is good reason to believe that the acid is not removed by the lime, and that the tubes after being washed in lime water should be baked at a temperature of 400 degrees to 450 degrees Fahrenheit for several hours. If carried out in this way the process of pickling would be a most beneficial one, as it not only removes injurious mill scale but also enables the inspector to detect many defective spots that would otherwise escape notice.

Answers received to letters of inquiry addressed by the writer to officers stationed at the principal navy yards and ship yards have led him to believe that the general practice in the Navy for the past ten years has been not to remove the mill scale.† During that time it has been the practice to provide a rather elaborate system of brass internal-feed pipes which with the iron form a galvanic couple almost as strong as copper tubes would. Zinc plates provided in the usual way with rough iron hangers very imperfectly fitted to either the plates or to the rusty boiler braces have, in the opinion of the writer, little effect in preventing galvanic action, although most beneficial for another reason which will be alluded to later on. It may be concluded, then, that our boiler tubes have suffered from galvanic action, and that this can be prevented by removing the mill scale from the water side of the tubes or by using only seamless tubes which are thoroughly pickled during the process of manufacture and

---

* Metcalf says: "Acid if not taken off completely will continue to act upon and rot the steel." "The power of acid can be illustrated. A lot of watch-spring steel is finished. * * * When last pickled the baking was neglected; the steel is tough, it hardens well and when tempered it is springy and strong. * * * In three or four weeks the spring maker begins to work on it. He reports that it is rotten and worthless."

†Mill scale may be removed from the outside of the fire tubes by a sand blast, and from the inside of water tubes by inserting a rod in each tube, putting a lot of such tubes in a rumble with sand, and revolving the rumble.

have no scale left on when completed, and by substituting galvanized steel or iron for brass internal feed pipes.

It is to be understood that the practice of pickling the boiler plates and braces which has prevailed for some years will be continued.

Assuming the possibility of free HCl being liberated in a boiler containing sea water, and also assuming that galvanic action might be set up, the writer made the following experiments to discover which would stand better under those conditions, viz: lap-welded steel or knobbled hammered charcoal iron. The steel was Bessemer of the grade called "welding steel," viz: .07 carbon, and it may be assumed that all lap-welded steel boiler tubes are of about the same composition, for unless the metal is pure and soft the tubes cannot be welded with any success. Eight samples, four of 3-inch tubes and four of 2-inch tubes, all No. 13 B.W.G., each about 5 inches long, were taken and accurately weighed and measured. Each 3-inch steel was connected to a 3-inch iron sample by a copper wire, No. 20 gage; the 2-inch steels were likewise connected to 2-inch irons. Then a pair of 3-inch steel and iron and a pair of 2-inch steel and iron were placed in one earthenware crock and the remaining pairs in another crock. The crocks were filled with a 2 per cent. solution of HCl (1.2 specific gravity) and left in a room which was heated about 70 degrees Fahrenheit during the day but not at night, when, it being winter, the temperature fell sometimes to freezing. Each week the samples were all removed, washed and weighed separately, reconnected as before with the same wire and replaced in the crock, and enough 2 per cent. solution HCl added to make up for loss by evaporation. When it was observed at the end of any week that the action had been slight, the crock was emptied and filled with perfectly fresh 2 per cent. solution. Since the acid from start to finish became gradually neutralized by uniting with the metal, it may be assumed that the average strength of the solution acting during the whole course of the experiments was about equal to a 1 per cent. solution. At the end of eleven weeks it was found that the copper wires had become so brittle that they could not be used, and the experiment was brought to

a close. It was then found that the iron samples had lost .406, .919, .296 and .810 grammes per square inch, while 1.754, 1.373, 1.143, 1.412 was lost by the steel, a mean loss of .607 grammes per square inch for the iron and 1.420 grammes per square inch for the steel. The differences are so great as to show that under the conditions of the experiment the lap-welded steel is far more soluble than the iron.

TABLE No. I.

Samples.	Number of sample.	Material.	Diameter. Inches.	Gage B. W. G. No.	Area. Sq. inches.	Loss in grammes per sq. inch.	Average.
No. 1 end.	1	Iron	2	13	73.84	.406	.6080
	4	Iron	3	13	109.87	.810	
	5	Steel	2	13	74.09	1.754	1.5830
	8	Steel	3	13	109.77	1.412	
No. 2 end.	2	Iron	3	13	110.98	.919	.6075
	3	Iron	2	13	74.34	.296	
	6	Steel	3	13	106.36	1.373	1.258
	7	Steel	2	13	74.32	1.143	

Samples united by wire in pairs as follows: 1 and 5; 4 and 8; 3 and 7; 2 and 6.

4. *The Use of Sea Water.*—Sea water is known experimentally to be a better solvent of steel and iron than fresh water. The concensus of opinion based on experiment is that boiler steel and iron corrode about equally in cold sea water, but that boiler steel corrodes more rapidly than iron in hot sea water. The fact that steel boilers last as long as iron ones is, in the opinion of the writer, due to the boiler shell being much cooler than the tubes, while the furnaces, combustion chambers and tube plates are always protected by a thin scale. The scale which forms on tubes generally forms unevenly and cracks off in places, so that it makes more difference whether the tubes are of steel or iron than it does about the boiler plates. As previously stated, however, a comparatively small proportion of sea water has been used, and, therefore, the steel tubes, although more soluble than iron, have probably not suffered appreciably more than the iron ones, and neither the one nor the other have been much damaged by the slightly brackish water except in so far as this has rendered galvanic action possible when other conditions

previously referred to were favorable. Fresh water is a very poor conductor of electricity, and if used exclusively galvanic action would be out of the question.

5. *Presence of Carbonic Acid in Water.*—It has been asserted by some writers—Thorpe, Greenwood and Turner—that the presence of carbonic acid (CO_2) is absolutely necessary before corrosion of iron in the water can ensue. A direct experiment, quoted by Spenmath, shows that this is not true, and that iron can be rusted when suspended in a closed vessel over a solution of caustic alkali which would inevitably absorb all the CO_2 present. It is true, however, that CO_2 in water materially aids corrosion.* Moreover, very little CO_2 is required, for the reason that it simply acts as a carrier to bring the oxygen into combination with the iron, itself not being used up in the operation. It is no doubt the corrosive agent in certain river waters which are noted for their injurious effects on boiler tubes, such as one of the tributaries of the Parana River, South America, the water from the mountain streams at Cape Town, South Africa, the juniper water at Norfolk, Va., from the Dismal Swamp, and the water from Loch Katrine, Scotland. The CO_2 is produced by the oxidation of animal and vegetable matter. The amount present in sea water is small and fairly constant, while the amount found in river water is greater and more variable.

A quantitative determination on ship board of CO_2 in the water to be used in the boilers would be quite impracticable, and the best way to avoid the injurious effects of that gas is to keep the water in the boiler always slightly alkaline by the use of NaHO (sodic hydrate or caustic soda), or else potassic hydrate (KHO, caustic potash), and frequent testing with red litmus paper. The writer does not consider that the test of boilers for acidity by blue litmus, as heretofore prescribed by the Navy regulations, has answered its purpose, since the water may be saturated with

*Any one can test the corrosive action of CO_2 for himself as the writer has done. Pour diluted acid over limestone in a bottle and lead the gas through a tube to the bottom of a glass vessel filled with distilled water, and place iron or steel in the water. In 24 hours very considerable rust will form, float to the top of the vessel and form a firm skin, like ice, over the surface. If this skin is broken it will quickly reform, completely covering the surface.

both O and CO_2, both very corrosive and yet not in sufficient quantity to affect the blue paper. Water procured from shore may be easily tested for organic matter by adding to it a little H_2SO_4. If organic matter is present it will be changed by the acid and turned dark. Since, however, in buying water for ships' use, one usually has only the choice between taking the particular kind offered or going without any, the proper way out of the dilemma is to provide every war vessel with sufficient evaporator power.

It is the opinion of the writer that a good deal of damage has been done at times to our boiler tubes by the CO_2 in the fresh water from shore, procured for the purpose of preventing the formation of scale on the heating surfaces.

6. *Oxygen Dissolved in Water.*—Thorpe states that "Nearly all natural waters contain oxygen in solution and can only be freed therefrom by prolonged boiling in vacuo." Spenmath states that water absorbs oxygen as follows:

At 32° Fahrenheit it will absorb 4.9 per cent. of its own bulk.

At 50° Fahrenheit it will absorb 3.8 per cent. of its own bulk.

At 68° Fahrenheit it will absorb 3.1 per cent. of its own bulk.

Stromeyer states that under 150 pounds pressure cold feed water absorbs 3.2 pounds of oxygen per ton. Assuming the most favorable conditions that the O unites with the Fe to form Fe_2O_3, then 3.2 pounds oxygen will unite with 7.2 pounds of iron, and if the boiler, say of the *Castine*, holds 12 tons of water there will be enough oxygen in it to completely corrode 38.4 pounds of steel, which would be equal to one and two-thirds tubes 2 inches outside diameter, 7 feet 10 inches long, and No. 13 B.W.G. thick. If FeO or Fe_3O_4 were formed the case would be still worse. Owing to the leaks about the boilers and elsewhere, the U. S. S. *Castine* generally used sixteen tons of make-up feed in seven days at sea. This would introduce enough oxygen to completely corrode two and one-quarter more tubes. If salt water were used for make-up feed the damage would be greater still. If a single tube leaked, probably half a boiler full of fresh water would be lost, and as evaporator power is not provided for such emergencies, salt water would have to be used.

In the writer's opinion the zinc used in our boilers has considerable effect in neutralizing the oxygen in the water. Since zinc is used as a coating for iron to prevent it from rusting, one might suppose that it would not rust in the boiler until all the iron had first become oxidized. This, however, appears to be far from the case. After zinc plates, rolled or cast, have been suspended in a boiler under steam for a few months, and no attention paid to making good electrical connections, the plates are often found to be quite brittle, with an earthy appearance.

Chemical analysis shows that the metallic zinc has been converted into oxide, consequently so much oxygen must have been taken up and prevented from uniting with the iron. The 3.2 pounds of oxygen in 1 ton of water would unite with 13 pounds of zinc, forming ZnO.

Although for many years there has been a regulation prohibiting the use of the boilers for trimming ship, a scrutiny of the log books shows very clearly that this regulation has not always been carried out. Then, again, few of our ships have any considerable tank space for storage of water for make-up feed, and it may be stated that the general practice is to use some of the boilers as fresh-water tanks, the water saturated with oxygen going in and out from time to time as required. Some chief engineers make a point of never lowering the surface of the water below the level of the tubes, and thus partially save the latter.

On the whole it may be concluded that much injury has been done to our boiler tubes by the oxygen in both fresh and salt water.

Recognizing the potency of the oxygen dissolved in water, either fresh or salt, as a factor in the corrosion of boiler tubes, the writer undertook the following experiments with a view to discovering what difference in corrodibility, if any, existed between the usual materials of which our tubes are made, viz: iron, Bessemer steel and open-hearth steel. Two samples of nickel steel (30 per cent. nickel) were tested at the same time to get an idea of the value of this metal for boiler tubes. The chemical and physical properties of the samples are given in the accompanying Table No. II.

TABLE No. II.—TABLE SHOWING LOSS OF WEIGHT OF SAMPLES OF STEEL AND IRON AFTER SIXTEEN WEEKS' EXPOSURE TO AIR AND DISTILLED WATER.

Number of sample.	Kind of material.	Outside diameter, inches.	Thickness B.W.G.	Total area. Square inches.	Chemical analysis. C.	P.	S.	M.	Sil.	Ni.	Physical properties. Ultimate tensile strength. Pounds per square inch.	Per cent. elongation in 8 inches.	Original weight. Grammes.	Weight after 16 weeks in bath. Grammes.	Loss. Grammes.	Loss per square inch area. Grammes.	Remarks.
1	30 per cent. nickel-steel	#.	15	56.4	.08	.05	.015	.39	...	.30	Not obtained.		263	260	3	.053	Bent hot.
2	" "	#.	15	56.4	.08	.05	.015	.39	...	.30	"	"	266	260	6	.106	Bent cold.
3	Seamless-drawn open-hearth steel tubes	3	8	40.88	.20	.010	.021	.40	.005	0	47,410	26.5	401	387	14	.342	These samples cut from one tube.
4	" "	3	8	43.18	...	...	...	...	...	...	47,410	26.5	414	401	13	.301	
5	" "	3	8	53.62	...	...	...	...	...	...	.....	.....	516	508	8	.149	These two cut from one tube.
6	" "	3	8	52.86	...	...	...	...	...	...	.....	.....	516	509	7	.132	
7	" "	3½	9	57.0	.11	.017	.026	.48	.04	...	55,000	28.0	461	444	17	.298	These two cut from one tube.
8	" "	3½	9	57.0	...	...	...	...	...	...	55,000	28.0	500	481	19	.332	
9	Lap-welded Bessemer steel tubes	3	9	57.4	.07	.100	.06	.35	.009	...	59,000	25.0	472	458	14	.244	All from one maker.
10	" "	3	9	57.0	...	...	...	...	...	...	.....	.....	479	467	12	.210	
11	" "	3	9	57.0	...	...	...	...	...	...	.....	.....	517	504	13	.228	
12	" "	3	9	57.0	...	...	...	...	...	...	.....	.....	472	459	13	.228	
13	Charcoal-iron tubes	3	9	51.72	trace	.034	.015	trace	...	(b)	47,000	25.0	439	426	13	.251	All from one maker.
14	" "	3	9	56.24	...	...	...	...	...	.80	.....	.....	466	452	14	.241	
15	" "	3	9	56.24	...	...	...	...	...	...	.....	.....	469	454	15	.266	
16	" "	3	9	57.06	...	...	...	...	...	...	.....	.....	490	475	15	.262	

a. Plate bent in form of segment of 3 inch tube.

b. Oxide and cinder.

Remarks on Appearance of the Surface.—The samples of seamless tubes were free from scale and annealed. The other samples all had mill scale left on them. Indications of pitting visible in every sample after sixteen weeks' exposure. Samples 3, 4, 5 and 6 from one maker. Samples 7 and 8 from another maker.

1. Few large rust spots on outside. About twice as many smaller ones on inside.
2. Few large rust spots on inside. About twice as many larger ones outside.
3. Outside all bright, except few spots of very thin scale about $\frac{1}{10}$ inch thick. Inside half bright and half rough scale thicker than that on outside.
4. Outside like No. 3. Inside like No. 3, except more rough scale.
5. Outside about ½ bright. Scale inside and outside hard and smooth.
6. Same, except outside about ¼ bright.
7. Outside about ⅛ bright. Scale outside hard and smooth, inside rough and flaky.
8. Outside about ⅛ bright, otherwise same as No. 7.
9. Outside about ¼ bright, balance covered with thin, smooth scale. Inside about ¼ bright. Rest covered with scale showing tendency to blister. Surface rougher than outside.
10. Outside about ¼ bright. Rest covered with thin, smooth scale. Inside about ¼ bright. Rest as No. 9.
11. Scale outside smooth and hard. Inside rough and flaky.
12. Same as No. 11.
13. Scale outside harder and smoother than inside.
14. Same as No. 13.
15. Similar to 13, but inside much rougher than 13.
16. Similar to No. 15.

The samples were segments of tubes about 6 inches long, except samples No. 1 and No. 2, which were flat plates bent to a radius of $1\frac{1}{2}$ inches, so that the jet of air would strike them in the same manner as it did the tubes. The samples were placed convex side up in an iron tank, porcelain lined, and were supported on glass, eight samples on each side, each quite separate from the others. Under each lot of eight ran a glass tube with openings of $\frac{1}{32}$-inch diameter at such intervals that one hole would be under the middle of each sample. The tank was kept filled with distilled water, and a continuous current of air delivered through the tubes and allowed to bubble up through the holes, the bubbles striking the middle of the inside of the sample and then flowing out to the ends. The air was furnished by a Richards air pump with blower attachment, such as is used for blow-pipe jets in laboratories. The samples were moved forward one place each week for sixteen weeks, so that at the conclusion of the test each piece had been in every position and consequently had been exposed to exactly the same influence as every other piece. At the end of that time the samples were removed, washed, weighed and photographed.

The first noticeable point about all the samples is the activity with which corrosion has gone on in such pure water in a comparatively short time. The second point is, that although the jet of air struck the center of the under (inner) side of the sample, in many cases the outer side is much corroded. Third, all of the samples show pitting more or less developed. This, however, was to have been expected in view of the statement of Rowan, that "pitting occurs in all kinds of corrosive liquids and all kinds of metals, even platinum." This shows how mistaken are those who look upon pitting as positive evidence of galvanic action, or who, like Mr. Dewrance (quoted by the London Engineer, March 9, 1900), ingeniously ascribe it to the formation of globules of HCl from the decomposition of $MgCl_2$.

In this connection it may be well to mention that four samples of $\frac{1}{8}$-inch O.D., 6 inches long, No. 17 B.W.G. cold-drawn nickel-steel (3 per cent. nickel) bicycle tubes were tested at the same time as the boiler-tube samples described in Table No. I. They

7. Outside about ½ bright. Scale outside hard and smooth, inside rough and flaky.
8. Outside about ½ bright, otherwise same as No. 7.
13. Scale outside harder and smoother than inside.
14. Same as No. 13.
15. Similar to 13, but inside much rougher than 13.
16. Similar to No. 15.

were placed in an earthenware crock by themselves in a 2 per cent. solution HCl. For the first four weeks the corrosion was very slow and uniform. Then the samples began to corrode, groove and pit, and were soon riddled with holes, the parts adjoining the holes being quite good.

Fourth, the progress of corrosion in the case of seamless-drawn tubes is quite peculiar. The oxide seems to form to a depth of about $\frac{1}{64}$ inch, then peel off, leaving the surface smooth and silver bright.

In figures 3, 4, 7 and 8, nearly the whole has peeled off the outside, while in figures 5 and 6 the oxide formed on the inside has only peeled in a few small spots, leaving the surface with a dark, smooth coat much like that on a new hot-drawn tube. Fifth. The avidity with which the oxygen has attacked the 30 per cent. nickel-steel is surprising, in view of the fact that a number of writers state that nickel-steel of about 25 per cent. nickel is very difficult to corrode, and one (Vosmaer) states that it is "practically incorrodible." Referring to the table, it will be observed that the loss of weight by the nickel-steel was far less than by any of the others. That two samples of the seamless-drawn tubes were exceptionally good, but the average of the other four samples, viz: 3, 4, 7 and 8, was much the same as the four lap-welded steel and four iron. It will be necessary to continue these experiments until about half of the samples corrode all the way through before an opinion can be formed as to the relative value of seamless and lap-welded tubes exposed to the air and moisture.

Assuming that the loss of weight per square inch of area for eleven weeks would have been $\frac{11}{16}$ of the loss for sixteen weeks, and since the samples 9 to 16 in Table No. II were furnished about the same time and by the same manufacturer as samples 1 to 8 in Table No. I, we may make the following comparison to show the relative loss of weight when exposed in one case to moist air and in the other case to 1 per cent. solution HCl and galvanic action.

Before leaving the subject of the effect of air in water, attention should be called to the fact that sea water contains air, and if boiled in an evaporator this gas must go over with the steam,

TABLE No. III.

No. of sample from Table I.	Metal.	Actual loss exposed eleven weeks to HCl and galvanic action, grammes per square inch.	No. of sample from Table II.	Metal.	Estimated loss if exposed eleven weeks to air and water, grammes per square inch.
1	Iron.	.406	9	Iron.	.159
2	Iron.	.919	10	Iron.	.146
3	Iron.	.296	11	Iron.	.170
4	Iron.	.810	12	Iron.	.180
	Average.	.607		Average.	.164
5	Steel.	1,754	13	Steel.	.167
6	Steel.	1,373	14	Steel.	.144
7	Steel.	1,143	15	Steel.	.168
8	Steel.	1,412	16	Steel.	.156
	Average.	1,420		Average.	.159

get cooled down in the condenser, churned up and thoroughly mixed by the air pump, which delivers the mixture into the feed tank. Sea water also contains sea weed, jelly fish and other organic matter, sufficient to be decomposed into quite an appreciable amount of CO_2, which also goes over with the air. Although oxygen, as stated before, can only be completely removed from water by boiling in vacuo, much of it goes off when the water is heated to a temperature of 175 degrees Fahrenheit to 190 degrees (Howe). The same may be said of the CO_2, and the obvious conclusion is that the temperature of the feed tank should be kept up to nearly the boiling point at all times, at sea and in port, by admitting live steam if necessary.

General Conclusions.—The tubes of our boilers may be made to last longer in the future than they have done in the past if engineers will be governed by the following rules, which have heretofore been "more honored in the breach than in the observance." Allow no tube or plate to enter the construction of the boiler with mill scale on the side in contact with the water. Use galvanized iron or steel instead of brass for internal feed pipes. These may rust out and should be made easily renewable. Provide evaporator power to ensure a sufficient supply of fresh water, so that it will never be necessary to use either sea water or fresh water from shore of doubtful quality. Provide fresh water reserve

tanks, so that the boilers will never have to be used as storage tanks. The reserve tanks to be provided with only two pipes, one to let the water in from the evaporators and the other for drawing the water out. An injector should be fitted for drawing the water out to prevent the fresh water from being accidentally pumped overboard by opening the wrong valve. The injector should deliver into the feed tank only. Keep all boilers closed, the water at constant level and above the tubes and, by use of caustic soda or caustic potash, at all times sufficiently alkaline to turn red litmus paper blue. Use zinc plates of the standard size, 12 inches by 6 inches by 1 inch, distributed beneath the surface of the water. These may assist in using up the oxygen and the experiment should be tried. Lippincott states that zinc is dissolved in caustic alkali with the following reaction:

$$Zn + 2\ KHO = K_2\ Zn\ O_2 + 2\ H.$$

If this is true then the nascent H might very well seize upon any free O in the vicinity, especially under the influence of the high temperature of the boiler. The experiment need not be continued long, for if after some months it was found that the zinc did not oxidize that would be a proof that it was useless. Finally keep the temperature of the feed tank up to nearly boiling at all times.

After all this has been done by the engineer there still remains the problem for the tube manufacturer to furnish tubes of some material less corrodible than iron or steel, and above all to produce tubes more uniformly corrodible. At present a lot of tubes are put into a boiler side by side, under identical conditions in every respect. One of them will corrode through in six months and another in six years.

If tubes cannot be made to wear more uniformly, then there should be some method of sorting them out, so that when put into a boiler a given lot could all be depended upon to last two or three or four years, according to their quality. Perhaps the art of photomicrography will help tube makers to attain this end. Certainly the manufacture of lap-welded tubes, as carried on at present, is in a very crude state, and much improvement would be made if the manufacturers could be brought to realize the

importance of the subject. The writer has been unable, as yet, to get sufficient data from which to conclude whether or not the seamless tubes will last longer or corrode more uniformly than the others.

All of the remarks heretofore made have applied to the tubes used in our naval boilers of the Scotch, low marine and locomotive marine types carrying not over 160 pounds of steam. It was considered worth while to discuss this subject because we have many such boilers and it will be a number of years before they are entirely superseded by water-tube boilers. With the advent of the water-tube boiler several new and serious difficulties arise, due to the fact that they are generally designed to carry a much higher pressure than the other types. There is reason to believe that at a temperature of above 356 degrees Fahrenheit, corresponding to a pressure of 146 pounds, iron or ordinary carbon-steel begins to decompose water. This would indicate the necessity of abandoning the use of those metals. It looks just now as if nickel-steel (25 per cent. nickel) was the most available material notwithstanding its great cost. The experiments tried by Yarrow, quoted by the "London Engineer," July, 1899, show that this metal when exposed to the action of the flame loses about one-third as much of its weight as is lost by mild steel under the same conditions. When exposed to the action of superheated steam it loses about one-half as much as mild steel does. The experiments previously described as made by the writer show that when exposed to air and moisture 30 per cent. nickel-steel loses about half as much as the best samples of mild steel and one-fourth as much as the average of the others. Since the flame, the superheated steam and the oxygen in the water are the only unavoidable agencies which destroy the tube it is evident that a nickel-steel tube can be depended upon to last at least twice as long as a low-carbon steel or an iron tube.

As its coefficient of expansion is about the same as brass (88 copper, 10 tin, 2 zinc), nickel-steel used in the boilers before mentioned might give trouble by leaking at the fire-box ends as did brass tubes. There would, however, be no trouble on this score when used in water-tube boilers, and that is where such material is especially needed.

With regard to the probable cost only a rough estimate can be made. With the current price of metals (July, 1900) the raw materials for brass boiler tubes (composition: 88 copper, 10 tin and 2 zinc) would cost about 18 cents per pound, and for 25 per cent. nickel-steel 15 cents per pound. The labor of manufacture of nickel-steel tubes would be greater than in the case of brass, since it is a very hard, tough metal. The percentage of defective tubes in every lot would at first, no doubt, be much greater than in the case of brass tubes. This is liable to be a source of considerable expense and a most uncertain one.

In those water-tube boilers like the Niclausse, in which the tubes do not have to be expanded, it might be possible to use steel tubes the surfaces of which have been case hardened, since it has been observed that the cutting edge of a cape chisel found in a boiler remained intact while the remainder was badly rusted.

In the same class of boiler it might be practicable to protect the surface next to the water by enameling with material similar to that now used extensively for the so-called "agate" kitchen ware, all of the ingredients required for the manufacture of this enamel being abundant and cheap.

It is not, however, for the engineer to encroach upon the province of the manufacturer or of the inventor, but only to point out to them the problems which they are to solve, and then leave the matter in their hands, hoping that they will apply themselves to this work promptly, diligently and successfully, and not force us to give up our water-tube boilers, with their higher steam pressure, just as we are beginning to fully appreciate their merits.

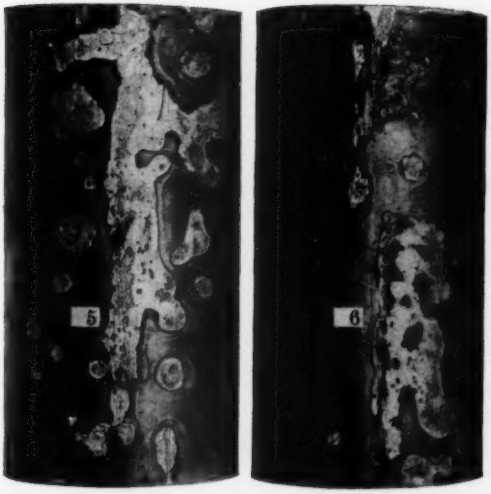
List of authorities referred to: 1. Shock, "Steam Boilers," 1880. 2. Stromeyer, "Marine Boiler Management and Construction," 1893. 3. Howe, "The Metallurgy of Iron and Steel," 1891. 4. Rowan, "Boiler Incrustation and Corrosion," 1895. 5. Spenmath, "Protective Coverings for Iron." 6. Thorpe, "Dictionary of Chemistry." 7. Comey, "Dictionary of Chemical Solubilities." 8. Lewis, "Service Chemistry," 1889. 9. Metcalf, "A Manual for Steel Users," 1900. 10. Greenwood, "Steel and Iron," 1893. 11. Lippincott, "Encyclopedia of Chemistry," 1880. 12. Turner, "Metallurgy of Iron and Steel," 1895. 13. Vosmaer, "Mechanical and other Properties of Iron and Steel," 1891. 14. Newman, "Metallic Structures," 1896. 15. Spons, "Workshop Receipts," 1896.

FIG. 1.

FIG. 2.

FIG. 3.

FIG. 4.



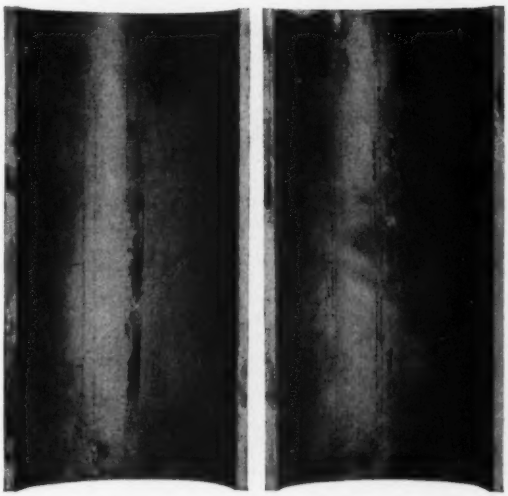


Fig. 5.

Fig. 6.

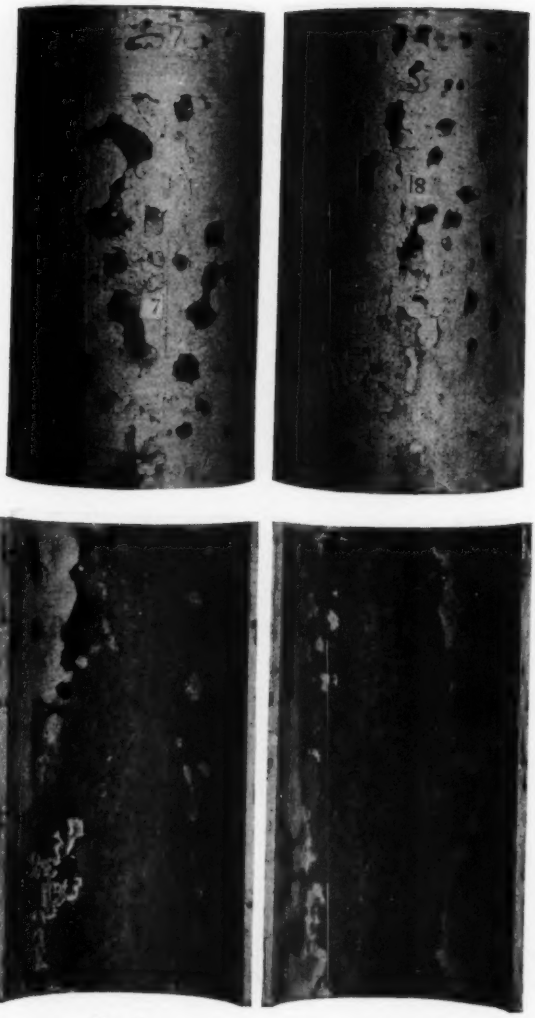


FIG. 7.

FIG. 8.

FIG. 9.

FIG. 10.

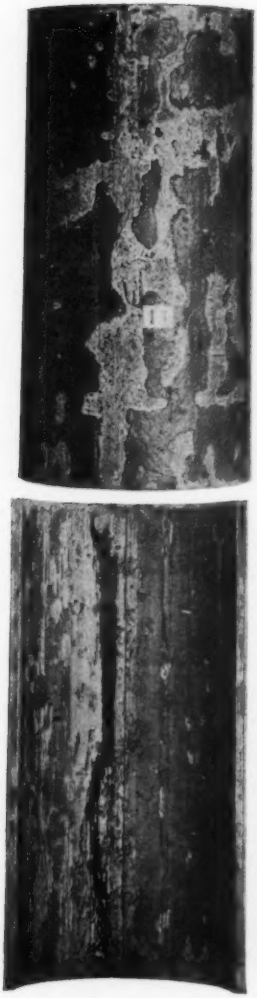


FIG. 11.

FIG. 12.

FIG. 13.

FIG. 14.

FIG. 15.

FIG. 16.

INERTIA STRESS OF ELASTIC GEARS.

By J. H. Macalpine, Esq., Associate Member.

[Continued from page 384, Vol. XII.]

Chapter VI.

THREE ILLUSTRATIONS OF FRICTIONALLY RESISTED MOTION. ENERGY TRANSFER.

1st Illustration.

22. MASS MOVING UNDER ELASTICITY OF SPRING ALONE.

I will not yet make the restriction that f is small, but will begin by giving three simple illustrations which will help towards a clear grasp of the whole phenomenon. Then I shall treat the general theory and deduce from it the effect when the restriction spoken of above is made. This will lead to an extremely simple graphical determination of the transfer of energy to or from the valve under these limiting conditions.

1st Illustration.—Suppose, when the engine is at rest, that the valve displaced by a force greater than the friction F. The subsequent motion till the valve comes to rest can at once be determined.

Let OX, Fig. 9, be a time base from which valve displacements are measured, and let the valve be displaced by a distance Oa. Draw O_1Y perpendicular to OX, taking $O_1a_1 = Oa$. Lay off a_1b_1 parallel to OX making $a_1b_1 = f \times O_1a_1$. Then a_1b_1 will represent the force required to displace the valve to a when there is no friction. And the straight line $b_1O_1b_2$ will give, by its abscissae measured from O_1Y, the elastic force of the gear for any displacement. Thus for the displacement cc', the elastic force is c_1c_2. When the valve is released at a it will begin to move towards OX and will be opposed by the friction F. If we lay off c_2c_3 to represent this force F, the resultant force urging the valve towards OX will be c_1c_3. If, now, through c_3 we draw d_1d_2

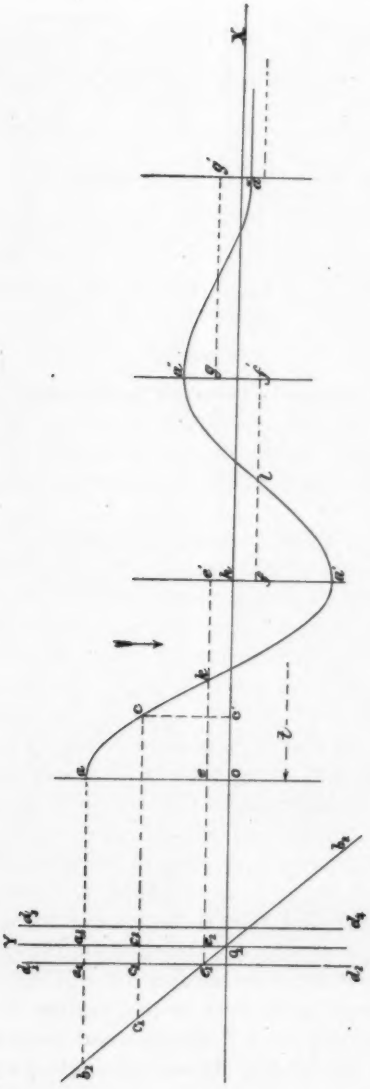


Fig. 8.

parallel to OY, the abscissae of $b_1O_1b_2$, measured from d_1d_2 will obviously give the resultant force for the corresponding displacement ordinate, *so long as the valve is sliding in the direction indicated by the arrow.* The straight line $b_1O_1b_2$ cuts d_1d_2 at e_1 and we have

$$e_1e_2 = F = f \times O_1e_2, \text{ or } O_1e_2 = \frac{F}{f}$$

Also, the resultant force is seen to be proportional to the displacement of the valve from ee', where $Oe = O_1e_2$. Thus, as we saw before, we have the same conditions as for unresisted motion with the exception that the center of motion instead of being OX is ee'. As the rate of increase of stress with change of strain is the same as for unresisted motion the time of passing from one extremity, a, of the travel to the other, a', will be unchanged and the motion will be represented by

$$y = + \frac{F}{f} + ae \cos \omega_s t.$$

The motion is thus continued till $\cos \omega_s t = -1$, when

$$y = + \frac{F}{f} - ae,$$

or

$$ae = a'e';$$

then, if the displacement of a' from OX is greater than F/f, the return movement at once begins, the friction being reversed at the same moment. The resultant force is now proportional to the abscissae of $b_1O_1b_2$ measured from d_3d_4 at a distance F from O_1Y, but on the other side from d_1d_2. The motion is now given by

$$y = - \frac{F}{f} + a'f \cos \omega_s t,$$

the displacement of ff' below OX being F/f.

The half travel will again be completed in the same time $T/2$, and when a'' is reached the amplitude of the vibration will again be suddenly cut down from $a''f' = a'f$ to $a''g$; or, by the same amount, $2F/f$, as before. The motion will thus continue with

the same diminution of amplitude at each reversal till a point a''' is reached, the distance of which from OX is less than F/f.

This illustration is introduced principally to exhibit clearly the discontinuous change which takes place in the curves at each reversal of the motion, and also to show that friction if constant in intensity has no effect on the natural period. If vibrations such as those figured here are recorded the true natural period would be got experimentally, at least in gears working a single valve. Where there are, as in the *Newark's* L.P. gear, two valves worked by one gear, the case is not so simple.

Since $\cos \omega_1 t$ appears as the variable in the expression of each part of the motion, aa', $a'a''$, . . . , there is no change from zero epoch at the reversals.

23. MEASUREMENTS OF ENERGY FOR 1ST ILLUSTRATION.

The changes of energy of the valve, which we will have to consider more fully later, are readily followed in Fig. 9. The total energy required to displace the valve, without friction, from O to a—that is, the whole potential energy of the strain at a—is represented by the area of the triangle $O_1a_1b_1$, and is equal to

$$\tfrac{1}{2}\, O_1a_1 \times f.O_1a_1 = \tfrac{1}{2} f \times O_1a_1^2 = \tfrac{1}{2} f \left(ae + \frac{F}{f}\right)^2. \quad . \quad (46)$$

This is, of course, equal to the kinetic energy that, were there no resistance, the valve would possess when passing OX, which would then be its center of motion; and at that moment the energy would be all kinetic.

The potential energy of the vibration aa' is less than this, being represented by the triangle $e_1a_2b_1$, equal to

$$\tfrac{1}{2} f \times ae^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

This measures the kinetic energy with which the valve passes the actual center of the vibration aa', a portion of the potential energy, expression (46), having been absorbed by friction.

When a' is reached the whole energy is again reduced to potential energy of the strain, equal to

$$\tfrac{1}{2} f \times ha'^2 = \tfrac{1}{2} f \left(ae - \frac{F}{f}\right)^2. \quad . \quad . \quad . \quad . \quad . \quad (48)$$

The total loss of energy is given by the difference of expressions (46) and (48),

$$= \tfrac{1}{2} f \left\{ \left(ae + \frac{F}{f} \right)^2 - \left(ae - \frac{F}{f} \right)^2 \right\} = F \times 2\, ae.$$

This is equal, as it obviously must be, to the work done against friction in passing from a to a', since $a'e' = ae$.

The fall from the potential energy of the vibration aa' to that of the vibration $a'a''$, is from $\frac{1}{2} f \times ae^2$, given by expression (47), to

$$\tfrac{1}{2} f \times a' f^2 = \tfrac{1}{2} f \left(ae - \frac{2F}{f} \right)^2.$$

Thus the loss of potential energy of vibration is

$$\tfrac{1}{2} f \left\{ ae^2 - \left(ae - \frac{2F}{f} \right)^2 \right\} = F \times 2 \left(ae - \frac{F}{f} \right) \quad . \quad . \quad . \quad (49)$$

If ae is very large compared with F/f, this expression is practically equal to $F \times 2ae$ lost in friction between a and a'.

I have deduced the loss of energy of vibration in this form, equation (49), as in the more general theory we must first determine the sudden change of amplitude of vibration at the reversal of the valve, and from that the loss of energy. But this does not present the clearest view of the subject. From the non-agreement of equations (46) and (47) it might appear that all the energy was not energy of vibration. But expressing the potential energy of the vibration aa' at the end of the stroke, when the valve itself has actually no energy, merely measures the kinetic energy it will have at the middle position k when some loss due to friction, which has still to take place when the valve is at a, has been deducted. Also, the difference between the loss of energy, equation (49), and the frictional loss $F \times 2ae$ in sliding from a to a' might at first sight appear as a contradiction of the conservation of energy. But this is not so. For the expressions used in equation (49) being those for the actual kinetic energy of the valve at the mid positions, k and l, of the vibrations aa' and $a'a''$, their difference should be exactly equal

to the frictional loss between those mid positions. And this is the case, for the factor $2(ae - F/f)$, equation (49) is equal to

$$ae + \left(ae - \frac{2F}{f}\right) = a'e' + a'f.$$

2d Illustration.

24. 1ST PERIOD AND SYNCHRONIZING VALUATIONS ONLY.—ACTUAL EPOCHS EACH ZERO.

Suppose the valve motion to consist only of vibrations of the 1st and the natural period, the engine being run at such a speed as will give exact synchronism.

Let us have

$$r = a_1 \cos \omega t - a_s \sin \omega_s t = a_1 \cos \omega t + a_s \cos\left(\omega_s t + \frac{\pi}{2}\right). \quad . \quad (50)$$

The actual vibration due to the synchronizing term has an amplitude a_s' at the time t, where

$$a_s' = B_o + \frac{a_s \omega_s t}{2}.$$

(See equation (15 d), in which, for the present illustration, I have supposed $A = o$).

Suppose the arbitrary constant, B_o, so large that we need not take particular account of the change of a_s' with the time during any one vibration except in so far as it affects the transfer of engery.

In the actual valve motion the synchronizing vibration will fall back quarter a period from the epoch given by equation (50), as shown in §15. We will thus have,—

$$r_1 = -\frac{F}{f} + a_1' \cos \omega t + a_s' \cos \omega_s t, \quad . \quad . \quad . \quad (51)$$

when r_1 is increasing.

$$a_1' = a_1/(1 - T^2/T_1^2).$$

The actual 1st period and synchronizing vibrations, equation (51), have each zero epoch.

The motion is illustrated in Fig. 10. The full line curve *abc* represents the one period of the vibration $a_1' \cos \omega t$, XX being a

time base and $t = 0$ at YY_1. The full line curve $a'b'c'$ represents the vibration $a_s' \cos \omega_s t$. The dotted lines ee, e_1e_1, e_2e_2, . . . are removed from XX by the distance F/f, and each part of the vibration $a'b'c'$ is laid off from the corresponding dotted line as a center line. Thus the total displacement of equation (51) at any time t is made up of the sum of the displacement of those portions of the curves abc and $a'b'c'$ each measured from the axis XX.

If we neglect the variability of a_s', $t = 0$ gives r_1 a maximum. The valve which has been moving out will now return if, as we will suppose, the elastic force is stronger than the opposing friction. Calling the displacement (from $t = 0$ till the next reversal of motion) r_2, we have

$$r_2 = +\frac{F}{f} + a_1' \cos \omega t + a_s' \cos \omega_s t + A \cos (\omega_s t - \lambda). \quad . \quad (52)$$

To determine the arbitrary constants A and λ we have the two conditions,—1st. The displacement cannot change suddenly on reversal of motion. 2d. The motion is then zero.
Or,—

$$\text{For } t = 0 \begin{cases} r_1 = r_2. \\ \dfrac{dr_2}{dt} = 0. \end{cases}$$

The first condition gives

$$-\frac{F}{f} = +\frac{F}{f} + A \cos \lambda, \text{ or } A \cos \lambda = -\frac{2F}{f}.$$

The second gives (still neglecting the variability of a_s') $A \sin \lambda = 0$.

Hence $$\lambda = 0 \text{ and } A = -\frac{2F}{f}.$$

Thus, equation (52) becomes

$$r_2 = +\frac{F}{f} + a_1' \cos \omega t + \left(a_s' - \frac{2F}{f}\right) \cos \omega_s t. \quad . \quad . \quad (53)$$

An examination of Fig. 10 shows that the variability of a_s' would have to be very great before this result would be sensibly changed.

There is no change of epoch of either vibration. But there is a sudden fall in the amplitude and energy of the synchroniz-

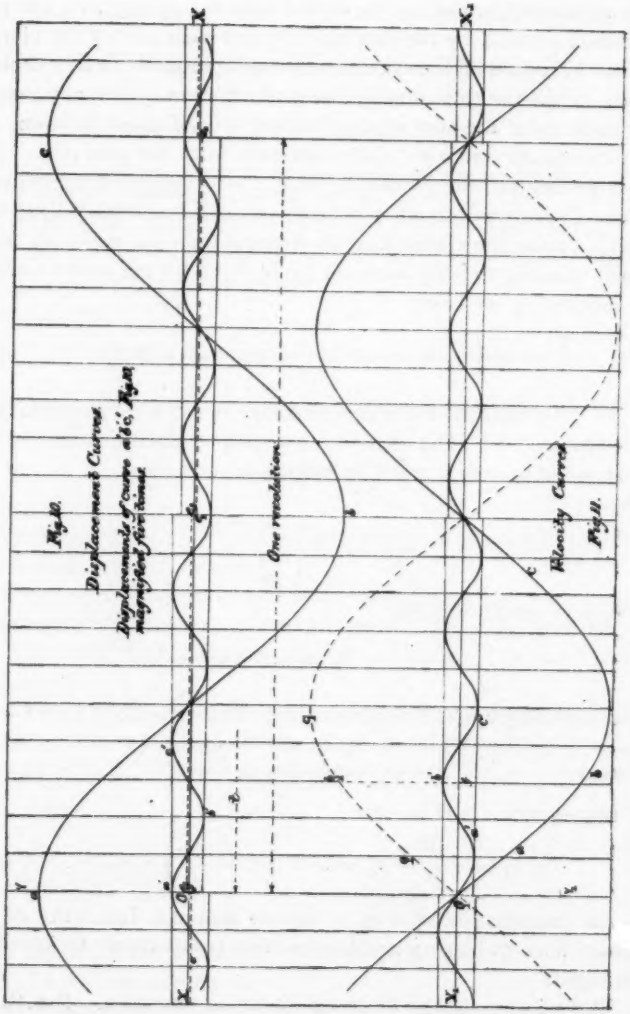


Fig. 10.
Displacement Curves.
Displacements of Curves a'b'c', Fig. 10, magnified five times.
One revolution.
Fig. 11.
Velocity Curves.

ing vibration. This does not mean that some energy has been destroyed, which would, of course, be impossible. As shown in the *1st Illustration*, the total energy at YY_1 is measured by the potential energy of the strain of the gear, the energy of the valve being zero. In passing YY_1,—that is changing from the conditions given by equation (51) to those given by equation (53)—the sudden fall of amplitude of the synchronizing term and increase of the constant term show that part of the energy, which in equation (51) was credited to the synchronizing vibration, remains in equation (53) as potential energy of strain of the gear till the motion given by the latter equation is again reversed.

As the actual synchronizing vibration is still quarter a period later in phase than that for the rigid gear, the conditions for communication of energy, and consequent growth of its amplitude, are unchanged. Thus the amplitude at any time t, so long as equation (53) holds, is

$$B_0 + \frac{a_s \omega_s t}{2} - \frac{2F}{f}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

In Fig. 11, *abc* and $a'b'c'$ are the velocity curves corresponding to the displacement curves *abc* and $a'b'c'$, respectively, of Fig. 10. The dotted curve $a_1b_1c_1$ is the curve *abc* turned around the base X_1X_1 as axis of rotation. Hence the intersection of the two curves $a'b'c'$ and $a_1b_1c_1$ shows that the velocities due to the two vibrations are equal and opposite; that is, that the valve is at a stationary point. These intersections only fall on the ordinates a, b, and c, of Fig. 10. Hence, for the amplitudes represented in the figures, equation (53) holds from a to b. The synchronizing period supposed in the figures is the 5th. For this, or any other odd period, the circumstances at ordinate b, Fig. 10, will be similar to those at ordinate a; that is, the maximum displacements of both vibrations occur at those times and they are in the same direction. At b, then, there will also be a sudden fall equal to $2\ F/f$ in the amplitude of the actual synchronizing vibration, but as there is still no change of epoch the growth of amplitude continues.

25. CHANGE OF a_s' PER REVOLUTION. LIMITING VALUE OF a_s.

At YY_1 the amplitude was—equation (54)—reduced to

$$B_s - \frac{2F}{f}$$

but in each complete vibration of the actual synchronizing vibration—in which $\omega_s t$ increases by 2π—equation (54) shows that it has grown by

$$\frac{a_s}{2} \times 2\pi = a_s\pi.$$

In one complete revolution of the engine the total increase would be $a_s s\pi$, as s is the number of synchronizing vibrations per revolution. But at both b and c there is a reduction $2F/f$. Thus on the whole there is a change of amplitude per revolution

$$= a_s s\pi - \frac{4F}{f}. \quad \ldots\ldots\ldots \quad (55)$$

If this is negative the synchronizing vibration will die out; or rather cannot appear. If positive, the amplitude will continue increasing till in Fig. 11 velocity curves $a'b'c'$ and $a_1b_1c_1$ have other intersections than those shown, and the rate of absorption of energy rises, due to more than a single reversal of the motion of the valve near each of the points a, b, c, Fig. 10.

This will not occur till the stress due to the actual synchronizing vibration is several times that due to the 1st period vibration, as may be readily shown thus,—

The maximum ordinate fb' of the velocity curve $a'b'c'$, Fig. 11, will be nearly equal to the ordinate fb_1 of the curve $a_1b_1c_1$ when this occurs.

We have for the velocity at b' [three-quarters of a period of this term after YY_1, which gives $\omega_s t = \frac{3}{4} \times 2\pi$], by differentiating the last term of equation (53),

$$\frac{d}{dt}\left(a_s' - \frac{2F}{f}\right)\cos\omega_s t = -\left(a_s' - \frac{2F}{f}\right)\omega_s \sin\omega_s t = \left(a_s' - \frac{2F}{f}\right)\omega_s a_1 b'.$$

If $s = 5$ the corresponding velocity b_1 of the first period is, from equation (53),

$$\frac{d}{dt} a_1' \cos \omega t = -a_1' \omega \sin \omega t = a_1' \omega \sin \tfrac{4}{5} \times \frac{2\pi}{5} (atb_1) = .81 a_1' \omega.$$

As the velocities are equal and opposite

$$\left(a_s' - \frac{2F}{f}\right)\omega_s = .81 a_1' \omega.$$

The ratio of the maximum stress is,

$$\frac{\left(a_s' - \frac{2F}{f}\right)\omega_s^2}{a_1' \omega^2} = \frac{\left(a_s' - \frac{2F}{f}\right)\omega_s}{a_1' \omega} \times \frac{\omega_s}{\omega} = .81 \times 5 = 4.05,$$

as $\omega_s/\omega = s = 5$, in Fig. 10.

The result would have been but little different had s been any odd number higher than five. Thus, for the relation of epoch of the two vibrations supposed in this illustration the change of amplitude per revolution would be given by expression (55) until the growth of the actual synchronizing vibration was such that the total inertia stress had risen to nearly five times that due to the 1st period. The valve would be in violent vibration and practically allowable limits would have been far exceeded. Hence, the limit of the admissable rigid gear amplitude of the synchronizing vibration, when equation (50) expresses the valve motion, is given by equating expression (55) to zero, thus

$$a_s = \frac{4}{\pi} \frac{F}{sf} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

s and f can be calculated from the drawings of the gear; and I shall show that, at least for the Marshall gear, a_s can be readily estimated with ample accuracy. But I will not stop to discuss the practical application as equation (56) is derived only from a particular case.

The conditions of stability of the epoch of the synchronizing vibration in the valve motion, supposed in this illustration, are discussed in the *2d Proposition*, Chapter VII.

3d Illustration.

26. 1ST PERIOD AND SYNCHRONIZING VIBRATIONS ONLY. ACTUAL EPOCHS ZERO AND 180°, RESPECTIVELY.

Suppose the epoch of the synchronizing vibration changed one half period from that shown in Fig. 10.

The motion for a rigid gear is then

$$r = a_1 cos\ \omega t + a_s sin\ \omega_s t = a_1 cos\ \omega t + a_s cos\left(\omega_s t + \frac{3\pi}{2}\right). \quad (57)$$

The synchronizing vibration falls back one quarter period and equation (51) is replaced by

$$r_1 = \pm \frac{F}{f} + a_1' cos\ \omega t - a_s' cos\ \omega_s t, \quad . \quad . \quad . \quad (58)$$

the sign of the constant term depending on the direction of motion of the valve, being positive if r_1 is decreasing.

We have $$\frac{dr_1}{dt} = -a_1' \omega\ sin\ \omega t + a_s' \omega_s sin\ \omega_s t.$$

The value of this velocity when t is very small is—putting $t = \delta t$—

$$-a_1 \omega\ sin\ \omega \delta t + a_s' \omega_s sin\ \omega_s \delta t = (-a_1' \omega^2 + a_s' \omega_s^2)\ \delta t,$$

replacing the sines of the small angles by the angles themselves. The terms $a_1' \omega^2$ and $a_s' \omega_s^2$ are the maximum accelerations due to the 1st and synchronizing elastic gear vibrations, and thus we see that the sign of the constant term F/f, in equation (58), will be positive, for δt positive, if the acceleration (or stress) from the 1st period is greater than the acceleration (or stress) from the actual synchronizing vibration. If either the sign of δt or the relative magnitude of the stresses is changed, the sign of F/f will also change. The meaning of this is clear from Figs. 12 and 13, which are in every way similar to Figs. 10 and 11. When $t = 0$, at YY_1, the displacements of the two curves, in Fig. 12, are opposite. Hence in Fig. 13 the tangents at the initial point O_1, drawn to the curves $a'b'c'$ and $a_1 b_1 c_1$ are inclined in the same way to $X_1 X_1$. The inclination of the tangent at any point of the velocity curve to $X_1 X_1$ measures the rate of change of velocity; that is, measures the acceleration, which is of course propor-

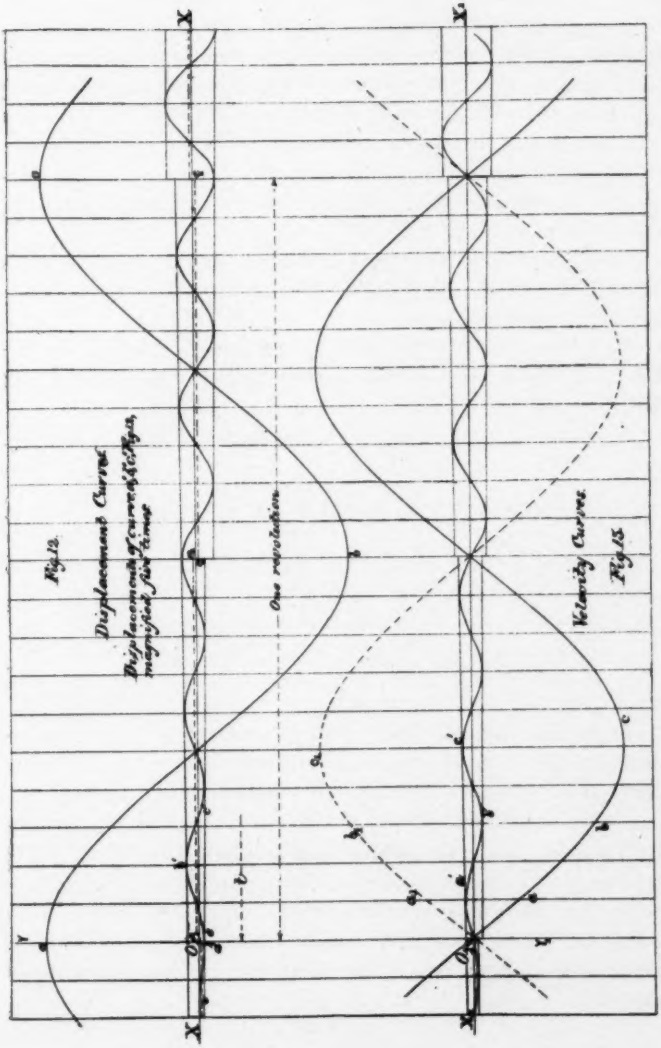


Fig. 12.
Displacement Curves
Displacement Curves of Figs. 13, magnified 500 times
One revolution
Fig. 13.
Velocity Curves

tional to the stress. If the tangent at O_1 to the curve $a_1b_1c_1$ is more inclined to X_1X_1 than that of the curve $a'b'c'$, Fig. 13, the negative velocity of the 1st period at a point immediately after O_1 is greater than the positive velocity of the synchronizing vibration. Hence the valve begins to move toward XX; that is, it moves inward under the difference of the forces due to the two periods, that of the 1st period being the greater. The center e_1e_1 will thus be above XX. Similarly in approaching YY_1 the center ee is below XX.

If the actual synchronizing vibration so increases that it gives a greater stress than that due to the 1st period the positions of the centers will be reversed.

Should the difference of forces on reaching YY_1 be less than F there will evidently be a momentary rest of the valve, as we found in the case of Fig. 8; but that we need not stop to consider.

Just before YY_1, if the 1st period stress is the greater, we have,

$$r_1 = -\frac{F}{f} + a_1' \cos \omega t - a_s' \cos \omega_s t. \quad . \quad . \quad . \quad (59)$$

The actual 1st period vibration has 0° epoch and the actual synchronizing 180° epoch.

Precisely as in the *1st Illustration* we find for r_2

$$r_2 = +\frac{F}{f} + a_1' \cos \omega t - \left(a_s' + \frac{2F}{f}\right) \cos \omega_s t. \quad . \quad . \quad (60)$$

The friction instead of absorbing the energy of the synchronizing vibration actually increases it; and as there is no change of epoch and no further reversal of motion till b is reached, the augmentation of amplitude proceeds.

97. INCREASE OF AMPLITUDE FOR REVOLUTION.

At b and c the circumstances of the motion are similar to those at a, if the synchronizing vibration is of odd period, and further sudden increase of amplitude by $2F/f$ takes place at each point. Hence over the whole revolution there is a total increase of amplitude

$$a_s 8\pi + \frac{4F}{f}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

This increase of the synchronizing vibration will quickly bring up its stress to equality with that of the first period, if, as we shall find to be the case, this relation of coincident and opposite maximum displacements are stable.

When the stress from the synchronizing vibration has become the greater the circumstances at O change, and the curves of velocities $a'b'c'$ and $a_1b_1c_1$ must have at least two other intersections near O_1. The epoch of the actual synchronizing vibration will change suddenly at this new point of reversal and we cannot follow the motion till we have made a more general investigation. But the curves of energy transfer, Fig. 21, will show that as the stress of the synchronizing vibration rises more and more above that of the 1st period vibration, the communication of energy per revolution arising from those sudden changes of amplitude quickly ceases; and on further rise still there is a rapidly increasing absorption of energy.

28. POSSIBLE LARGE INCREASE OF INERTIA. STABILITY OF EPOCH.

Without careful investigation it would be impossible to say that the circumstances of Fig. 12 would not be nearly fulfilled in any particular engine. Hence—excepting those cases in which we shall find the viscosity of the gear to play an important part—we must always count on the inertia stress being more than double that deduced from the valve motion diagram. The communication of energy through the frictional resistance is, indeed, a curious, but far from being an isolated phenomenon. We are reminded of the excitation of the violin string by means of the friction of the bow, although the phenomena are by no means identical.

The consideration of the stability of this relation of coincident and opposite maximum displacements of the two vibrations will be given in the *1st Proposition*, Chapter VII. It will be shown that so long as the stress from the actual synchronizing vibration is lower than that from the rest of the valve motion the arrangement in Fig. 12 is stable. Thus, if the conditions of the *3d Illustration* are once fulfilled, there will be no hindrance, except viscosity of the gear, to the continued growth of this term

till the total inertia stress has been at least doubled. Indeed, the communication of energy by friction in this case has nothing to do with the existence of a synchronizing term in the rigid gear motion; that is, expression (61) is still finite ($=4\,F/f$) when $a_s = 0$. Hence the vibration of natural period will grow to importance if the time intervals between the maximum displacements a and b, b and c, Fig. 12, are very approximately an odd multiple of half the natural period, even when no rigid gear synchronizing vibration exists, or unless the growth is prevented by the opposite tendency of the existing synchronizing vibration.

The diminution of the coefficient of friction with increase of velocity of sliding, shown by equation (26), will also be seen to foster the growth of this short period vibration.

If the synchronous vibration figured in Figs. 10 to 13 had been of even period and its displacement at a in the same direction as that of the 1st period, as in Fig. 10, the displacement at b would have been opposite, and the sudden decrease of amplitude F/f, at a, would have been compensated by an equal sudden increase at b. The only change of energy over the complete revolution would be due to the constant growth of amplitude given by equation (23). But as in the actual case the simple curve of sines abc is replaced by a more complex curve the maximum ordinates Oa and e_1e_2b are not usually separated by exactly half a revolution. Thus the circumstances of either the *2d* or *3d Illustration* may be practically reproduced whether the period of the synchronizing vibration is even or odd.

29. CAUSES OF ENERGY TRANSFER AT REVERSAL EXPLAINED. TRANSFERS IN 2D AND 3D ILLUSTRATIONS. MULTIPLE REVERSALS.

It will be useful to explain here the curves in Figs. 21 and 22 and this will supplement the partial view of the subject given in the foregoing illustrations. If F is very small compared with the maximum inertia stress of the actual synchronizing vibration, as supposed in these figures, the change of epoch of this term at reversal is, as we would expect, small. The exact expression for the maximum change of epoch we will investigate later. The curves of Figs. 21 and 22 show the transfer of energy

due to friction when for a single revolution the change of epoch and the change of amplitude are neglected; that is, they give very approximately the transfer due to friction in one revolution for the mean epoch and mean amplitude of the actual synchronizing vibration during that revolution.

In the *2d Illustration*, when $t = 0$, the amplitude of the actual synchronizing vibration suddenly falls from a_s' to $a_s' - 2F/f$. As the rigid gear synchronizing vibration is one quarter period ahead it is then on its center of motion, and the total stretch of the gear due to this period is thus, at this time, measured by the amplitude of the actual vibration. The energy is then all potential energy of the strained gear and—see equation (46)—suddenly changes from

$$\tfrac{1}{2} f a_s'^2 \text{ to } \tfrac{1}{2} f \left(a_s' - \frac{2F}{f} \right)^2.$$

The energy deducted from the vibration being the difference of these two quantities

$$= 2Fa_s' \left(1 - \frac{F}{fa_s'} \right).$$

As fa_s' is the force required to draw out the gear by a_s',—or approximately, the maximum inertia of the actual synchronizing vibration—it is by supposition much larger than F. Thus we may write for the sudden decrease of energy, the simpler expression $2Fa_s'$. At *b*, Fig. 10, a similar decrease will take place on reversal. Hence, since we are actually considering a_s' as the mean value over one revolution, we may write for the total absorption of energy due to friction in that time

$$4Fa_s'.$$

Similarly, in the *3d Illustration* friction communicates energy to this amount per revolution.

In Figs. 21 and 22, energy supplied by friction is measured upward from XX, and that abstracted measured downward. The abscissae are epochs of the actual synchronizing vibration; thus the ordinate marked 0°, Fig. 21, corresponds to the con-

dition in the *2d Illustration*, Fig. 10, the energy absorbed per revolution being

$$4Fa_s'.$$

The ordinate marked 180° corresponds to the conditions of the *3d Illustration*, Fig. 12, and the energy supplied per revolution is

$$4Fa_s',$$

so long as the maximum stress from the actual synchronizing vibration is not greater than that from the 1st period.

The curves for Fig. 21 are drawn for the synchronizing term being of the 5th period, supposing the curve of displacements for all the other terms to be a sine curve, as in Figs. 10 and 12, where *abc* follows this law; or, as already explained, if *abc* is the actual curve the circumstances of reversal at *a*, *b*, and *c*, are to be exactly similar. If the period is odd but higher than the 5th the curves would be practically the same.

The curves of Fig. 22, are for an even period synchronizing term, or opposite circumstances of reversal at *b* from those at *a* and *c*, Figs. 10 and 12.

Let us take first the curves Fig. 21. In Fig. 10, if we moved the curve $a'b'c'$ along to the left till b' fell on the ordinate Oa, the conditions at YY_1, Fig. 12, would be reproduced and the actual synchronizing vibration would have changed its epoch by 180°; the energy transfer would also have changed from absorption to communication. For any intermediate change of epoch the change of rate of energy transfer will be intermediate. This is what is shown in Fig. 21. If the motion of the valve is so nearly frictionless that we have practically zero change of epoch at reversal, when the actual synchronizing vibration is still so small that its maximum stress is a very small fraction of that due to the other terms expressing the actual valve motion, we will get the curve *aa*, which is a curve of sines.

When the actual synchronizing vibration rises to give a maximum stress equal to that from the other terms we have the curve marked 1.0, there being a cusp at the ordinate 180°. Energy is supplied to the valve at the same rate as before at this ordinate,

but the rate of supply is quickly affected by change of epoch and the range of epoch over which the valve has energy supplied is much restricted. The curves are also drawn for the cases in which the stress of the actual synchronizing vibration is 1.1, 1.2, 1.4, 1.6—3.4 times that of the stress due to the other terms.

In the curves above 1.0, the point on ordinate 180° is one, first of minimum communication of energy, quickly passing as the stress increases to one of maximum absorption.

Near this ordinate on each side, but at an increasing distance as the stress rises, there is a singular point on the curves forming a *point saillant*, since two branches meet here at a finite angle. At this epoch there is a sudden change in the number of reversals, as we shall see when considering the method of obtaining the curves. If the curves for still higher stress were drawn there would be other sudden changes in their nature at each change in the number of reversals, and the rate of absorption of energy would rise for every value of the epoch.

We have already seen in the *2d Illustration*, § 25, that for the conditions in Figs. 10 and 11 (0° epoch, Fig. 21,) there will only be one reversal near each end of the valve travel till the maximum stress from the actual synchronizing vibration is about four times that from the 1st period (in the actual case from all non-synchronizing terms at the end of the travel). As Fig. 21 only gives curves up to 3.4, the singular points do not come to the 0° epoch. In the *3d Illustration*—see § 27—single reversals only hold till the maximum stress of the actual synchronizing vibration equals that from the 1st period. This also agrees with Fig. 21, as at epoch 180°, the curve marked 1, and all higher, have singular points.

In any actual valve motion, between two successive passages, in opposite directions, through the center of motion there must be 1, 3, ... , reversals—that is, an odd number. The names *single reversal*, *triple reversal*, ... *multiple reversal*, are applied according to the number of reversals.

Fig. 22 is got from Fig. 21. The ordinate at 20° or 160°, Fig. 22 is half the sum of the 20° and 160° ordinates, Fig. 21, and thus we get curves symmetrical about the 90° ordinate. This

is due to the supposed opposite circumstances at the two extremes of the valve travel.

The ordinates give the value of τ, the coefficient of energy transfer due to friction, so that the energy absorbed (τ positive) per revolution is

$$\tau F a_s'.$$

Thus in Fig. 21, for the curve 2.4 and 140° epoch of the actual synchronizing vibration, $\tau = +7.1$, so that the energy absorbed per revolution is

$$7.1 F a_s'.$$

These curves really only apply to the case in which the valve curve *abc*, Figs. 10 and 12, is a sine curve. For ordinarily, as we have seen in *Examples I and II* for the *Newark*, the stresses will be different at *a* and *b*; in which case that of actual synchronizing vibration cannot be taken in the same ratio to each. If then for any epoch of the actual synchronizing vibration, similar circumstances held at *a* and *b*, but the ratios of the stresses due to the two curves were, say, 1.1 and 1.4, the ordinate of the exact energy transfer curve would at this epoch be about the mean of those of Fig. 21, marked 1.1 and 1.4.

Also, as the distances between *a* and *b*, *b* and *c* are actually different we may have any intermediate variety between the curves of Fig. 21 and those of Fig. 22.

For any particular valve gear we shall find that the proper energy transfer curve can readily be deduced from those of Fig. 21.

Chapter VII.

General Theory of Motion Resisted by Constant Friction.

30. General Change of Motion at Reversal.

A more general view of the theory of the motion can now be taken.

I will suppose exact synchronism, though the results would be of a nearly similar nature if the synchronism were only approximate.

If equation (1) gives the motion with a rigid gear the actual valve motion, so long as there is no reversal, and r_1 is increasing, is given by

$$r_1 = k - \frac{F}{f} + \Sigma_1^{\infty} a_n' \cos(n\omega t - \varepsilon_n) + \left(B_0 + \frac{a_s \omega_s t}{2}\right) \sin(\omega_s t - \varepsilon_s) + A \cos(\omega_s t - \lambda), \quad . \quad . \quad (62)$$

where the synchronizing term is taken from under the summation sign and given the same form as equation (15d).

If r_1 is decreasing the only change in the form of equation (62) is a change in the sign of F. Hence, all through what follows, the effect is of reversal of the valve motion from increasing to decreasing displacement; but to obtain the result of reversal from decreasing to increasing displacement it is only necessary to change the sign of F.

At reversal I will suppose the elastic force to be greater than F, so that there is no finite rest of the valve. The only case of finite rest which is practically important is that already treated, F being large.

The displacement r_2 after reversal is,—

$$r_2 = k + \frac{F}{f} + \Sigma_1^{\infty} a_n' \cos(n\omega t - \varepsilon_n) + \left(B_0 + \frac{a_s \omega_s t}{2}\right) \sin(\omega_s t - \varepsilon_s) + A_1 \cos(\omega_s t - \lambda_1). \quad . \quad . \quad (63)$$

Suppose reversal occurs when $t = t_r$. Then the displacements r_1 and r_2 are equal and, as there can be no finite change of velocity, that of both r_1 and r_2 are zero. We have,

$$t = t_r \begin{cases} r_1 = r_2, \\ \dfrac{dr_1}{dt} = \dfrac{dr_2}{dt} = 0. \end{cases}$$

The first condition gives

$$-\frac{F}{f} + A\cos(\omega_s t_r - \lambda) = +\frac{F}{f} + A_1 \cos(\omega_s t_r - \lambda_1),$$

or, $$A_1 cos\,(\omega_s t_r - \lambda_1) = -\frac{2F}{f} + A cos\,(\omega_s t_r - \lambda). \quad . \quad . \quad (64)$$

From the second and third conditions,

$$\frac{dr_1}{dt} = \frac{d}{dt}\left\{ \Sigma_1^{\infty}\, a_n' cos\,(n\omega t - \varepsilon_n) + \left(B_o + \frac{a_s \omega_s t}{2}\right) sin\,(\omega_s t - \varepsilon_s) \right\}_{t=t_r} - A\omega_s sin\,(\omega_s t_r - \lambda) = 0.$$

$$\frac{dr_2}{dt} = \frac{d}{dt}\left\{ \Sigma_1^{\infty}\, a_n' cos\,(n\omega t - \varepsilon_n) + \left(B_o + \frac{a_s \omega_s t}{2}\right) sin\,(\omega_s t - \varepsilon_s) \right\}_{t=t_r} - A_1\omega_s sin\,(\omega_s t_r - \lambda_1) = 0.$$

Whence, by substration,

$$A_1 sin\,(\omega_s t_r - \lambda_1) = A sin\,(\omega_s t_r - \lambda). \quad . \quad . \quad . \quad (65)$$

The value of A_1, obtained by squaring and adding equations (64) and (65), is

$$A_1^2 = A^2 - \frac{4FA}{f} cos\,(\omega_s t_r - \lambda) + \frac{4F^2}{f^2}. \quad . \quad . \quad . \quad (66)$$

In using this subsequently it must be carefully recollected that when the reversal is from a decreasing to an increasing displacement the sign of F is to be changed.

31. THE MOTION AND CHANGES AT REVERSAL REPRESENTED GRAPHICALLY.

1st Proposition.

Let us now represent the conditions (64) and (65) graphically. In Fig. 14, OX, OY, are two rectangular axes, the valve displacements being measured along OY. If in any gear we knew the epoch ε_s of the rigid gear synchronizing term we could, for any time t, lay off Oa at the angle $\omega_s t - \varepsilon_s$ to the negative direction of OX. Similarly, knowing a_s and B_o we could make $Oa = (B_o + a_s\omega_s t/2)$. Drawing ad perpendicular to OY will give $Od = (B_o + a_s\omega_s t/2)\, sin\,(\omega_s t - \varepsilon_s)$, one part of the actual synchronizing vibration. Similarly, for any assumed values of the arbitrary constants A and λ we can draw $ab = A$ and making the

angle $\omega_s t - \lambda$ with the positive direction of OY, positive angles being counted in the direction of the arrow. We have

$$Oe = \left(\frac{B_o + a_s\omega_s t}{2}\right) \sin(\omega_s t - \varepsilon_s) + A\cos(\omega_s t - \lambda),$$

the actual synchronizing vibration of the equation (62).

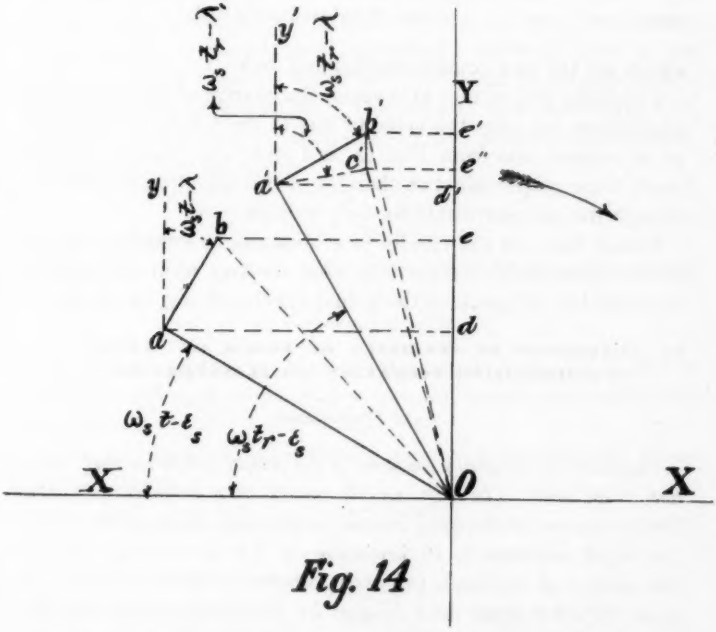


Fig. 14

We must now set the system Oab revolving about O in the natural period T, increase Oa at the rate $a_s\omega_s t/2$, the angle Oab remaining constant. The motion of e will be that due to the actual synchronizing vibration and as we know the rest of the valve motion the next time, t_r, of reversal can be obtained graphically from a diagram such as Fig. 11. Suppose it occurs at e', Fig. 14, the revolving system having reached $Oa'b'$. If the change is from r_1 increasing to decreasing we must now, from b',

draw $b'c' = 2F/f$ in the negative direction, parallel to OY. The new revolving system till the next reversal will be $Oa'c'$, and Oe'' will give the new value of the displacement of the actual synchronizing vibration. The angle $b'Oc'$ gives the change of epoch, and $a'c' = A_1$. For, drawing $a'y'$ parallel to OY we have,

$$a'c'\cos c'a'y' + b'c' = a'b'\cos b'a'y',$$

and

$$a'c'\sin c'a'y' = b'c'\sin b'a'y',$$

which are the two conditions (64) and (65).

Condition (65) is that at reversal the change of amplitude and epoch does not alter the velocity due to the synchronizing term, as is evident also from Fig. 14, and it is also obvious that we could trace graphically the changes of this vibration indefinitely, though the process would be very tedious.

I shall first use the results just obtained to establish two important propositions regarding what we may term the stability or instability of epoch of the actual synchronizing vibration.

32. CONDITIONS OF STABILITY OF EPOCH OF ACTUAL SYNCHRONIZING VIBRATION, 3D *ILLUSTRATION*.

1st Proposition.

Suppose the arbitrary constant of the actual synchronizing vibration to be zero. (In Fig. 14 Ob would then coincide with Oa.) Let a reversal of the valve motion occur when the revolving vector Oa, which indicates by its projection on YY the varying value of this term, is in its lowest (highest) position. The valve motion due to all the other terms must be near its maximum displacement upward (downward). So long as the stress from the actual synchronizing vibration at reversal is less than the maximum stress from the nonsynchronizing terms of the actual motion its epoch will be stable; that is, if a small arbitrary impulse changing the epoch is given, it will gradually return to its original value.

If we reckon $t = 0$ at the time of reversal the actual synchronizing term is

$$-\left(B_0 + \frac{a_s\omega_s t}{2}\right)\cos\omega_s t.$$

When Oa, Fig. 14, passes the negative part of the axis OY the motion of a is still downward, as the length Oa is growing. Hence the part of the displacement of the valve due to all the other terms, while very near a maximum, must still be increasing. The case now being considered is nearly that represented in Fig. 12, the actual valve curve being substituted for abc and the coincidence of the maximum displacements a and a' not being exact. For a short length near a we may replace the now complex curve abc by a portion of a curve of sines of the same curvature; that is, which will give the same displacements, velocities, and accelerations. The axis of this new curve will, of course, not usually be XX, Fig. 12, but is taken parallel to it. Thus, instead of the terms

$$k + \Sigma_1^{\infty} \frac{a_n}{1 - T^2/T_n^2} \cos(n\omega t - \varepsilon_n),$$

in equation (22), we may more simply write the equivalent curve of sines,

$$k_e + a_e \cos \omega_e (t + \delta t).$$

If the axis of the new sine curve changes from XX, k and k_e will have different values, and ω_e will not usually be exactly the same as ω.

Before reversal, if no arbitrary impulse is applied, the equation of motion is

$$r_1 = k_e + a_e \cos \omega_e (t + \delta t) - \left(B_o + \frac{a_s \omega_s t}{2}\right) \cos \omega_s t. \quad . \quad . \quad . \quad (67)$$

$$\frac{dr_1}{dt} = -a_e \omega_e \sin \omega_e (t + \delta t) + \omega_s \left(B_o + \frac{a_s \omega_s t}{2}\right) \sin \omega_s t - \frac{a_s \omega_s}{2} \cos \omega_s t. \quad . \quad (68)$$

$$\frac{d^2 r_1}{dt^2} = -a_e \omega_e^2 \cos \omega_e (t + \delta t) + \omega_s^2 \left(B_o + \frac{a_s \omega_s t}{2}\right) \cos \omega_s t + a_s \omega_s^2 \sin \omega_s t. \quad . \quad (69)$$

As reversal takes place at $t = 0$ equation (68) gives

$$-a_e \omega_e \sin \omega_e \delta t - \frac{a_s \omega_s}{2} = 0.$$

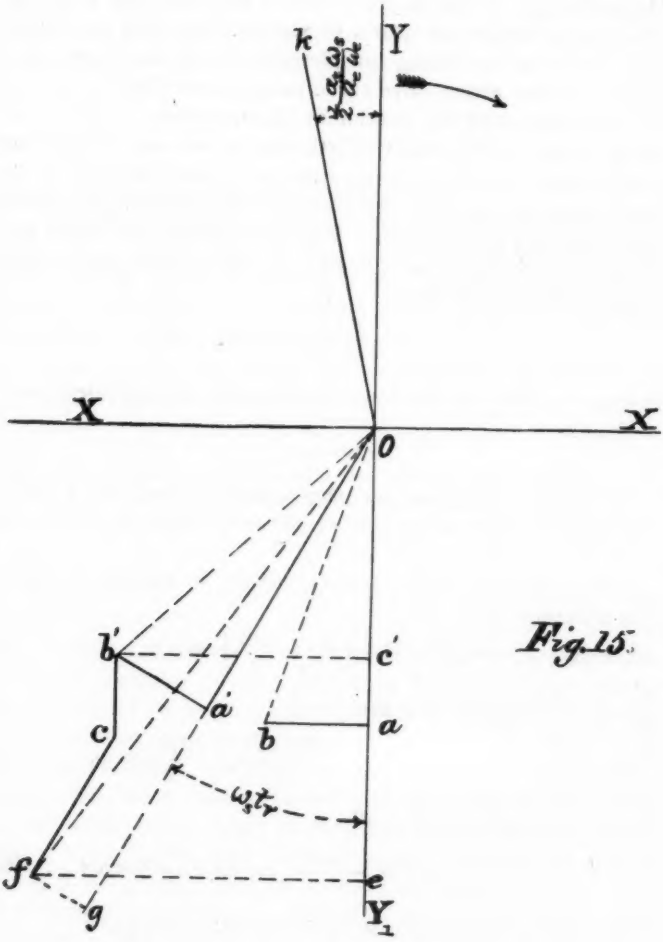


Fig. 15.

As $\omega_e \delta t$ is small

$$a_e \omega_e^2 \delta t = -\frac{a_s \omega_s}{2}, \text{ or } \omega_e \delta t = -\tfrac{1}{2}\frac{a_s \omega_s}{a_e \omega_e}.^*$$

Equation (67) becomes

$$r_1 = k_e + a_e \cos\left(\omega_e t - \tfrac{1}{2}\frac{a_s \omega_s}{a_e \omega_e}\right) - \left(B_o + \frac{a_s \omega_s'}{2}\right)\cos\omega_s t. \quad . \quad . \quad (70)$$

Suppose now a very small arbitrary impulse of amplitude A applied so as to change the epoch of the actual synchronizing vibration. In Fig. 15, Oa represents the revolving arm for this term in equation (67), when $t = o$; and ab, drawn at right angles to Oa, we will take as the revolving arm for the arbitrary impulse.

Should we suppose the impulse to have an arm not at right angles to Oa, then ab would be one component of this impulse and the component along Oa would merely make a slight change in the amplitude B_o to which no definite numerical value has been assigned. Hence the supposition made in Fig. 15 is quite general.

The arm Ok which, if drawn to scale would be many times longer than Oa, is that for the equivalent term of amplitude a_e and has an angular velocity ω_e. Its position as shown is for $t = 0$.

---

* For clearness let us estimate this value in the case of the *Newark's* L.P. valve gear, supposing it full ahead, as that will give the largest value of $\omega_e \delta t$. Suppose we are dealing with the maximum displacement near b, Fig. 1, where the inertia stress is only a little different from that for a simple harmonic motion of the same amplitude as the actual valve motion. There $\omega_e = \omega$ very nearly.

The maximum displacement is—See § 10—4.425 inches.

From Table X, $\frac{\omega_s}{\omega} = s = 6$ and $a_s = 10^{-3} \times 1.505$ feet.

Hence $\omega_e \delta t = -\tfrac{1}{2} \times \dfrac{10^{-3} \times 1.505 \times 12}{4.425} \times 6 = -.0122$ radian.

A radian being 57°.3, this gives,

$$\omega_e \delta t = .0122 \times 57.3 = .7 \text{ of a degree.}$$

Near a', Fig. 1, it can readily be shown that this angle will not have a greatly different value.

In an engine with a very stiff gear, which runs smoothly, $\omega_e \delta t$ will be much smaller than for the *Newark*.

The equation of motion before reversal now becomes,

$$r_1 = k_c + a_c cos\left(\omega_c t - \tfrac{1}{2}\frac{a_s\omega_s}{a_c\omega_c}\right) - \left(B_o + \frac{a_s\omega_s t}{2}\right) cos\ \omega_s t + A\ sin\ \omega_s t. \quad (71)$$

Fig. 15 is drawn for A positive, and the inclination of the arm Ob, which is now that of the actual synchronizing vibration, has added a positive vertical component to the velocity due to this term. The previous negative component due to the growth of ob parallel to the original revolving arm Oa still remains, but if the reversal is from increasing to decreasing displacement—that is, if the actual synchronizing vibration gives the smaller inertia stress at reversal, as supposed in the *3d Illustration*—evidently the whole system must revolve to a new position $Oa'b'$ before reversal now takes place, as the total velocity of the valve is now positive for $t = 0$, and the maximum upward displacement slightly postponed. If A is negative, or the actual synchronizing vibration gives the larger stress, the reversal is hastened.

From equation (71)

$$\frac{dr_1}{dt} = -a_c\omega_c sin\left(\omega_c t - \tfrac{1}{2}\frac{a_s\omega_s}{a_c\omega_c}\right) + \omega_s\left(B_o + \frac{a_s\omega_s t}{2}\right) sin\ \omega_s t - \left(\frac{a_s}{2} - A\right)\omega_s cos\ \omega_s t. \quad (72)$$

As t is very small at reversal, we may, in determining this time, t_r, from equation (72) write $cos\ \omega_s t_r = 1$ and replace the sines by the angles themselves. Thus,

$$-a_c\omega_c\left(\omega_c t_r - \frac{1}{2}\frac{a_s\omega_s}{a_c\omega_c}\right) + \left(B_o + \frac{a_s\omega_s t_r}{2}\right)\omega_s^2 t_r - \left(\frac{a_s}{2} - A\right)\omega_s = 0.$$

Neglecting t_r^2 this gives

$$(a_c\omega_c^2 - B_o\omega_s^2)\ t_r = A\omega_s,$$

or angle

$$a'OY_1 = \omega_s t_r = \frac{A\omega_s^2}{a_c\omega_c^2 - B_o\omega_s^2} \quad (73)$$

Equation (69)—since δt is very small—shows that the denominator in equation (73) is proportional to the difference of the inertia stresses due to the two terms at the original time of re-

versal, $t = 0$, before the arbitrary impulse A was given. Thus equation (73) bears out what has just been said regarding the change in the time of reversal due to the arbitrary impulse.

The arm is now suddenly changed from Ob' to Oc, $b'c$ being equal to $2\ F/f$ and drawn parallel to OY, downward if the stress of the actual synchronizing vibration is the smaller.

As the revolution proceeds c will move outward in the direction cf always parallel to the changing direction of Oa'. Still supposing the inertia stress from this term the smaller, the next reversal will take place in nearly half a revoltion of the engine, $\omega_s t$ will have increased by about $s\pi$, and cf to about $a_s s\pi/2$.

What is required is whether the angle $a'Of$ is numerically greater or smaller than $a'Ob'$.

Og, the projection of the broken line $Oa'b'cf$ on Oa', is

$$Og = Oa' + \frac{2F}{f}\cos \omega_s t_r + cf = Oa' + \frac{2F}{f} + cf,$$

since $\omega_s t_r$ is very small.

Similarly,

$$fg = A - \frac{2F}{f}\sin \omega_s t_r = A - \frac{2F\omega_s t_r}{f}.$$

Angle
$$a'Of = \frac{fg}{Og} = \frac{A - \frac{2F\omega_s t_r}{f}}{Oa' + \frac{2F}{f} + cf} \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

The denominator of equation (74) contains only positive terms. Hence the angle $a'Of$ is of the same sign as the numerator. Substituting for $\omega_s t_r$ from equation (73) we get for the numerator,

$$\frac{A\left\{a_c\omega_c^2 - \left(B_c + \frac{2F}{f}\right)\omega_s^2\right\}}{a_c\omega_c^2 - B_c\omega_s^2} \quad . \quad . \quad . \quad . \quad . \quad (75)$$

Thus if the stress of the actual synchronizing vibration for $t = 0$ is smaller than the maximum stress from the rest of the terms, and if the added amplitude, $2F/f$, does not change it to the larger, the angle $a'Of$ is positive for A positive. As it is obviously, from Fig. 15, algebraically less than $a'Ob'$ there is

what may be termed a nonoscillatory subsidence of the arbitrary disturbance, and the original epoch is stable.

When the maximum stress from the actual synchronizing vibration is greater than that from the rest of the terms, it can readily be shown by means of diagram like Figs. 11 and 14 (to determine the times of reversals and the effect) that if one of the reversals occurs as in the *1st Proposition* that there usually results a finite change of epoch after each set of reversals. The discussion of this case would be tedious and is not essential to our purpose.

If $a_e\omega_e^2 = B_o\omega_s^2$, expressions (74) and (75) become infinite. But they do not apply to this case as the rejected term involving l_r^2 would then be of importance.

The importance of this proposition is obvious on reference to the energy transfer curves, Fig. 21. For the conditions supposed are those corresponding to epoch 180°. At this ordinate, so long as the stress from the actual synchronizing vibration is the smaller, there is a maximum communication of energy, by friction, to the valve. This condition is liable to be fulfilled in every engine, and the proposition shows that if fulfilled there is no escape from the total inertia stress being doubled.

33. CONDITIONS OF STABILITY OF EPOCH OF ACTUAL SYNCHRONIZING VIBRATION, 3*D ILLUSTRATION*.

2d Proposition.

Suppose the arbitrary constant A of the actual synchronizing vibration to be zero. Let a reversal of the valve motion occur when the revolving arm Oa, Fig. 14, is in its highest (lowest) position, and the valve motion due to all the other terms near its maximum displacement upward (downward). It is required to find what are the conditions of stability of epoch of the actual synchronizing vibration.

This is the relation holding between the movements of the 1st and synchronizing periods in the *2d Illustration* and at ordinate 0°, Fig. 21, where we have a maximum absorption of energy.

Fig. 17 shows this case and is similar to Fig. 15.

The deduction is similar to that of the *1st Proposition* and need not be set down fully.

We find $$\omega_s t_r = \frac{A\omega_s^2}{a_o\omega_o^2 + B_o\omega_s^2} \quad \ldots\ldots\ldots \quad (76)$$

$$Og = Oa' - \frac{2F}{f} + cf.$$

$$fg = A - \frac{2F\omega_s t_r}{f}.$$

I shall make the angles $a'Of$ and $a'Ob'$ positive when b' and f are to the left of Oa' as in Fig. 17.

Angle $$a'Of = \frac{fg}{Og} = \frac{A - \frac{2F\omega_s t_r}{f}}{Oa' - \frac{2F}{f} + cf}.$$

The denominator is positive in all cases we need consider. The numerator is

$$\frac{A\left\{a_o\omega_o^2 + \left(B_o - \frac{2F}{f}\right)\omega_s^2\right\}}{a_o\omega_o^2 + B_o\omega_s^2}. \quad \ldots\ldots \quad (77)$$

This is positive, in every practical case, for A positive. Thus the angle $a'Of$ is of the same sign as $a'Ob'$, and the stability or instability of epoch depends on whether the former is less or greater that the latter.

Angle $$a'Ob' = \frac{a'b'}{Oa'} = \frac{A}{Oa'}.$$

Angle $$b'Of = \text{angle}\,(a'Ob' - a'Of) = \frac{A}{Oa'} - \frac{A - \frac{2F\omega_s t_r}{f}}{Oa' - \frac{2F}{f} + cf}$$

$$= \frac{\frac{2F\omega_s t_r}{f} \times Oa' - A\left(\frac{2F}{f} - cf\right)}{Oa'\left(Oa' - \frac{2F}{f} + cf\right)}.$$

The sign again depends only on the numerator, which on sub-

stituting $B_o + a_s\omega_s t_r/2$ for Oa', using equation (76), and rejecting A^2, reduces to

$$A\left\{cf - \frac{2F}{f}\,\frac{a_c\omega_c^2}{a_c\omega_c^2 + B_o\omega_s^2}\right\}. \quad . \quad . \quad . \quad . \quad (78)$$

If the actual synchronizing vibration has not grown beyond practically allowable limits—limits indicated in §25—the next reversal will take place in about half a revolution of the engine, when cf will have increased to about $a_s s\pi/2$. The numerator (78) will then become

$$A\left\{\frac{a_s s\pi}{2} - \frac{2F}{f}\,\frac{a_c\omega_c^2}{a_c\omega_c^2 + B_o\omega_s^2}\right\}. \quad . \quad . \quad . \quad . \quad (79)$$

Equation (56) gives the limit of the admissible magnitude of a_s and expression (79) is positive if a_s is equal to or larger than this limiting value. The epoch is then stable. Should the same conditions hold at the other extremity of the valve travel the actual synchronizing vibration will grow very large. For, as shown under the *2d Illustration*, the number of reversals does not increase till the maximum stress for this period has become nearly four times that from the rest of the terms. In most cases there is no escape from a breakdown of the gear.

If a_s is somewhat smaller than this limit the epoch will still be stable if $B_o\omega_s^2$ is large enough compared with $a_c\omega_c^2$. But, as we have seen in the *2d Illustration*, there will then be a progressive decrease of the amplitude a_s' and the epoch will soon become unstable.

34. DEFINITION OF *NORMAL EPOCH*, ORDINATE AND VECTOR.

Before proceeding we must make the following definitions:

Normal Epoch. If $A = 0$ in equation (62) the actual synchronizing vibration may be said to have its *normal epoch*; or, in other words, the actual synchronizing vibration has its *normal epoch* when it lags one quarter period behind the rigid gear term of the same period.

The actual synchronizing vibration, when $A = 0$,

$$= \left(B_o + \frac{A_s\omega_s t}{2}\right) sin\,(\omega_s t - \varepsilon_s) = \left(B_o + \frac{A_s\omega_s t}{2}\right) cos\left(\omega_s t - \varepsilon_s - \frac{\pi}{2}\right),$$

which is one quarter period behind $A_s\cos(\omega_s t - \varepsilon_s)$, the rigid gear term.

Normal Ordinate. The *normal ordinate* of the energy transfer curve is that ordinate which has the normal epoch for abscissa.

Normal Vector. The *normal vector* of the actual synchronizing vibration is the vector Oa of Fig. 14, for which this vibration has its normal epoch.

If Oa were constant, the motion of d, which is the projection of a on OY, would be simple harmonic. And if Oa, as figured, were the position of the revolving arm for $t = 0$, the epoch of the motion of d would be the angle aOY. But in the actual synchronizing vibration we have seen that the energy communicated by the gear causes Oa to lengthen at a uniform rate, without affecting its constant angular velocity ω_s. The vibration of d is no longer simple harmonic, but the angle aOY for $t = 0$, may still very properly be called the epoch of the vibration of d. Also, it follows that, when there is no friction the normal epoch will be retained if once established.

Consistently with the previous notation ε_s' is the epoch of the actual synchronizing vibration. Let us also put ε_s'' for the normal epoch of this vibration.

Then
$$\varepsilon_s'' = \varepsilon_s + \frac{\pi}{2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (79_1)$$

We may also put
$$\varepsilon_s' - \varepsilon_s'' = \alpha; \quad . \quad . \quad . \quad . \quad . \quad . \quad (79_2)$$

or, α is the angle by which the vector of the actual synchronizing vibration lags behind its normal vector.

35. TWO CONDITIONS THAT THE EVENTS IN SUCCESSIVE REVOLUTIONS MAY BE THE SAME.—1ST AND SYNCHRONIZING PERIODS ALONE PRESENT.

But this is not the case when there is friction. In Fig. 16, if the normal epoch obtains before reversal and Oa is the position of the normal vector at reversal, we have a sudden change from Oa to Oc, where $ac = 2F/f$ and is parallel to OY.

Energy has, in the case figured, been suddenly abstracted from the actual synchronizing vibration, since Oc is shorter than Oa;

but while the system aOc is revolving with the angular velocity ω_n an increase cf is added to the vector Oc at the uniform rate $a_1\omega_1 t/2$; cf always being parallel to the normal vector Oa, as they both revolve at the same rate. The increase of cf tends

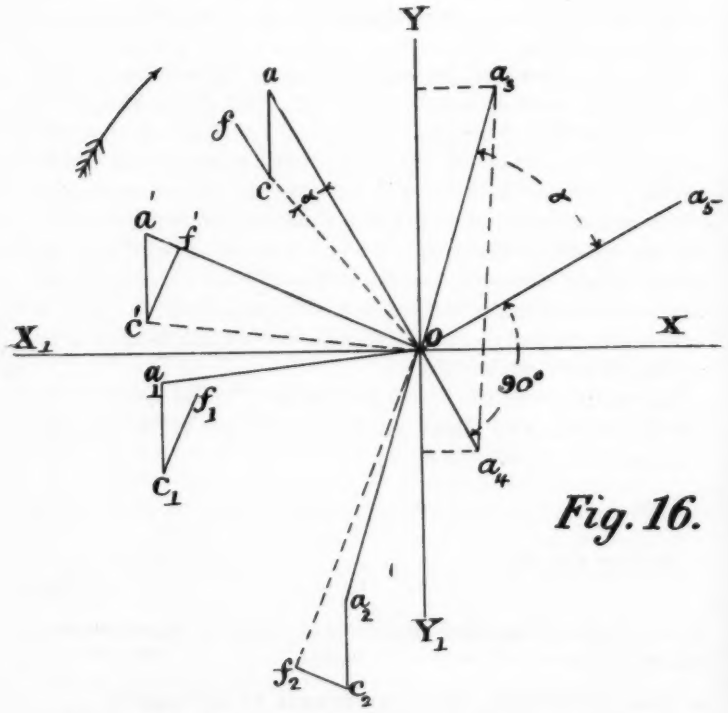


Fig. 16.

gradually to restore the normal epoch, but can never completely restore it; for, however large cf becomes, Of can never coincide with Oa. At next reversal there may be a further sudden departure from Oa, which will be followed by continued growth like cf, still parallel to Oa, and so on.

The important question arises,—What will be the limit of these progressive changes, so that the events in each revolution of the

engine will be an exact repetition of those in the preceding revolution?

Let us take again the simple case in which the rigid gear valve motion is made up of two simple harmonic motions; the longer one having a period of revolution equal to that of the revolution of the engine, and the shorter the natural period of the gear, so that exact synchronism is established. Let us also make the synchronizing term of odd period, so that the circumstances of reversal at each extremity of the valve travel may be the same.

Suppose that there is only a single reversal near each extremity of the valve travel. In Fig. 16 if Oa' is the vector of the actual synchronizing vibration at reversal we will have the sudden change $a'c'$ parallel to OY. There will be about $s/2$ revolutions of $Oa'c'$ before next reversal, Oc' being added to by a uniform growth like cf, parallel to the normal vector. This normal vector we may suppose to be revolving at any angular distance from Oa'.

The circumstances of the second reversal will not be the same as at Oa' unless the growth of the vector fulfils two conditions—

1st. The growth must be exactly along $a'c'$.

2d. It must, in half a revolution of the engine, be exactly equal to $a'c'$.

Stated otherwise, both the change of epoch and amplitude of the actual synchronizing vibration produced at any reversal must be annulled before the next reversal.

Since the growth due to the energy communicated by the gear would then be along $a'c'$, we see that the amplitude Oa' must be such that reversal occurs when the normal vector is passing OY; that is, when the actual synchronizing vibration, if of normal epoch, would have its maximum displacement.

If both conditions are not fulfilled, progressive change of epoch or amplitude, or both combined, cannot be arrested so long as the reversals near each extremity of the valve motion are only single. The second condition depends on an exact balance in which four independent quantities are involved; that is, we must have

$$\frac{a_s s\pi}{2} = \frac{2F}{f}.$$

The number s depends not only on the strength of the gear, but also on the mass, M, of the valve, and it is altogether unlikely that F, a_s, M, and f should have such values that the second condition would be complied with.

36. IN SIMPLE CASE, §35, Σ_s' MUST CHANGE TOWARDS 180° TILL TRIPLE REVERSALS OCCUR.

It may be logically concluded from the above that in this simple case, changes of epoch will continue till there is at least a triple reversal at each end of the valve travel; but as it is important to gain a perfectly clear grasp of the action it will be well to consider it a little further.

I will only indicate what may readily be expanded into a proof.

The epoch of the 1st period is taken as zero, the value it had in the *2d and 3d Illustrations*. Thus if $\epsilon_s' = 0$, reversal will occur when $t = 0$, as in the *2d Illustration:* then the vector Oa', Fig. 16—not the normal vector—is passing OY. It follows that if $a'OY$, at reversal, is a small angle, we know that ϵ_s' is near zero. Similarly for 180° epoch of the actual synchronizing vibration, reversal will happen when Oa' is passing OY', as in the *3d Illustration;* and if $a'OY'$ is a small angle at reversal ϵ_s' must be near 180°.

Suppose the growth cf per half revolution of the engine is greater than $2F/f$, or ac, Fig. 16. If the normal epoch ϵ_s'' is either 0° or 180° the two preceding propositions show that it is stable and growth of amplitude would proceed till the reversals were at least triple. If the normal ordinate is intermediate between 0° and 180° it can readily be proved by a diagram like Fig. 16 that while the normal epoch will be departed from, changes of epoch and amplitude will result in multiple reversals. This may be left to the reader.

It may be pointed out that for this rate of growth of cf, the limit of admissible magnitude of a_s, given in equation (56) for the conditions of the *2d Illustration*, has been exceeded.*

* If the reader returns to this case after studying the curves of change of epoch at reversal, Fig. 16_2, and the law of restoration of normal epoch in equation (84), he will find that the normal epoch will not be greatly departed from. Hence the growth of cf will be nearly added to Oa. The increase of Oa will then go on till the rate of absorption of energy will, near the normal ordinate, be greater than the value $4Fa_s'$,

If the growth is less than $a'c'$ we need only consider two cases.

1st. If reversal from increasing to decreasing displacement occurred at Oa', when $a'OY$ is less than 90° the only way of checking increase of epoch towards 180° would be for the growth $c'f'$ before next reversal to be towards a point between O and a', thus involving diminished energy of the actual synchronizing vibration. Now, the velocity due to the rest of the valve motion at reversal is not only opposite to that due to Oa', but is increasing—as we may see by drawing the corresponding velocity curves like Figs. 11 and 13—hence, if Of' with its smaller velocity had been substituted for Oa', the reversal would have occurred earlier, as then the velocity due to the rest of the valve motion would have been smaller. That is, it would have fallen back to such a position as Oa_1, when the growth c_1f_1 is no longer able to prevent retardation of phase occurring, if directed to a point between O and a_1.

2d. C_1f_1 may be directed to a point on Oa_1 outside a_1. This is shown at $Oa_2c_2f_2$. Here f_2c_2, less than a_2c_2, may still prevent increase of epoch towards 180°, but it obviously involves increasing energy of the actual synchronizing vibration. Continuance of this condition will, of course, lead to increase in the number of reversals. Any other case may be similarly treated.

Thus for any curve of Fig. 21, if the normal ordinate is between 0° and the next singular point, which indicates that there is only one reversal per half revolution, the epoch will change so that the ordinate to be measured will fall back towards the singular point. The recession will not stop even there, but I will show that on further recession this tendency will rapidly be weakened, and the growth cf, tending to restore the normal ordinate will further check it.

37. I_s MAY BECOME IMPORTANT THOUGH $a_s = 0$.

(An important inference can at once be drawn. Even if c_2f_2 were zero—that is, if there were no rigid gear synchronizing

shown for the 0° ordinate, Fig. 21—much greater, if cf is much greater than ac. Also, to obtain the requisite absorption of energy due to friction, Fig. 21 shows at once, the ratio I_s/I_e will be greater the nearer the normal ordinate is to 0°. But, even for normal epoch 180° the ratio cannot be much under 1.6, and the valve will be in serious vibration.

term—the arm Oa_2 would still be lengthened, and we can readily see that, whenever the conditions at both ends of the valve travel are favorable, a vibration of natural period will grow till its maximum inertia about equals that from the rest of the valve motion, and the total stress is doubled. The favorable condition is obviously that the time interval between the two extremes of valve displacement should be very nearly an odd multiple of half the natural period. As reversal must occur in this case when Oa_2, Fig. 16, is near OY_1—for only then will a_2c_2 add much energy to Oa_2—the ordinate at which energy transfer is measured must be near 180°, Fig. 21. Hence if the stress from this natural period vibration grew much larger than the limit just given, energy absorption would commence and the continued growth would be at once checked as there is no supply of energy from the gear.)

38. TWO CONDITIONS THAT EVENTS IN SUCCESSIVE REVOLUTIONS MAY BE THE SAME.—ACTUAL CASE.

In any actual case we will rarely have the conditions at the two ends of the valve travel the same. But it is obvious that two conditions of the same kind as those in §35 must hold if the events in one revolution are to be an exact repetition of those in the preceding revolution. These conditions are,—

1st. Over a whole revolution the total change of epoch of the actual synchronizing vibration must be zero.

2d. Over a whole revolution the total change of amplitude, or energy, of the actual synchronizing vibration must be zero.

39. CHANGES OF a_s' AND ε_s' SMALL, PER REVOLUTION IF J AND J' ARE SMALL.

If ac, Fig. 16, is a comparatively large fraction of Oa, we have a large change of epoch at reversal when the angle Oac is nearly a right angle; when this angle is small we have a large change of amplitude. For intermediate values of this angle we have a comparatively large change both of amplitude and epoch. To follow the events of reversal, change of amplitude, and change of epoch, through the whole revolution under a great variety of initial conditions would be very tedious and complex. The results would be further complicated if the growth cf were also so

large as to produce important changes of amplitude or epoch during one revolution.

The only practical case in which for τ positive we are in danger of serious growth of the actual synchronizing vibration (a_s'/a_s becoming large) is when the friction is much smaller than the rigid gear inertia of the valve at reversal. That is, when

$$J = \frac{F}{I_s} \text{ is small,}$$

where I_s is the rigid gear inertia at reversal, or at the maximum displacement in Fig. 1.

Unless I_n, the maximum value of the actual synchronizing vibration, is somewhat nearly as large as I_s no very serious overstressing of the gear will result, if designed to meet safely the rigid gear stresses. Hence we need only consider

$$J' = \frac{F}{I_s} \text{ small.}$$

Now in Fig. 16,

$$\frac{ac}{Oa} = \frac{2F}{f} \div a_s' = \frac{2F}{I_s}, \quad \ldots\ldots \quad (80)$$

if a_s'/a_s is large; for then obviously $fa_s' = I_s$.*

*Only when a_s'/a_s is large can we write $I_s = fa_s'$. As pointed out in § 15, the stretch of the gear due to the natural period vibration, is the difference of the two terms (23) and (24), when the actual synchronizing vibration has its normal epoch.

Let $Oa_3 = a_s'$, and $Oa_4 = a_s$, Fig. 16, Oa_4 is 90° ahead of the normal vector Oa_5. The projection of a_3a_4 on YY_1, obviously gives the stretch of the gear due to this period. If Oa_3 was constant the maximum stretch and inertia would occur when a_3a_4 was parallel to YY_1; but as it is growing, parallel to Oa_5, the maximum inertia and time of its occurrence are very slightly changed.

If Oa_3 has the normal epoch, coinciding with Oa_5, $a_3a_4 = a_s'^2 + a_s^2$; and with a_s' constant we would have $I_s = f\sqrt{(a_s'^2 + a_s^2)}$, which is little different from fa_s' if a_s'/a_s is large. The *Newark*, *Table X*, has values of a_s'/a_s ranging from 18.94 to 52.68, and the normal epoch is departed from only by angles of from 9° — 22′ to about 14° (25° in *Table X* in column for .76 cut-off being an over estimate, as explained). The expression for I_s in the text is thus a very close approximation in the case of the *Newark*, and evidently also for all cases where a_s'/a_s is large; and this approximation will hardly be affected if the growth of a_s' is taken into account, which the reader may readily do for himself.

The maximum errors will be, for a_s' constant, when Oa_3 lies along Oa_4. Then $I_s = f(a_s' + a_s)$ or $= f(a_s' - a_s)$; but as the actual synchronizing vibration would not then get any energy from the gear, the error will never approach this.

Thus for J' small we will have only small changes of amplitude and epoch at reversal.

We may also confine ourselves to cases where a_s does not greatly exceed the limit given by equation (56), as we have seen—see §36 and foot note—that whatever the epoch of the actual synchronizing vibration we would then have serious vibration of the valve.*

The growth of cf per half revolution is

$$cf = \frac{a_s s\pi}{2},$$

See §25.

Hence,
$$\frac{cf}{Oa} = \frac{a_s s\pi}{2a_s'} = \frac{fsa_s\pi}{2fa_s'} = \frac{fsa_s\pi}{2I_s}.$$

But if a_s has the value given in equation (56) this expression becomes

$$\frac{2F}{I_s} = J'.$$

Thus, by confining ourselves to J and J' small, we have only small changes of a_s'/a_s and of the epoch of the actual synchronizing vibration throughout one revolution, due either to ac or cf, Fig. 16; and we may, with good approximation, regard a_s' as constant during this period.

Chapter VIII.

Motion resisted by constant friction. j and J' small.

40. Curves of change of ε_s' at reversal.

We may now, without sensible error, study the total change per revolution of ε_s' and a_s', and also the transfers of energy due to friction and that communicated from the gear, supposing the reversals determined for the mean values of ε_s' and a_s'; that is, we may suppose ε' and a_s' to remain unaffected throughout the revolu-

* As this is true for every epoch in the simple law of valve motion supposed in §35, it will be true for any actual case, when usually a different normal epoch has to be taken at each end of the valve travel.

tion when determining the times of reversal. The whole question now becomes very simple.

I will first develop the method of drawing the curves of change of epoch at reversals.

The projection of $a'c'$, Fig. 16, normal to Oc' is

$$a'c' \sin Oa'c' = \frac{2F}{f} \sin a'OY,$$

and the angle $a'Oc'$ by which ε_s' is increased is very approximately

$$\frac{2F}{a_s' f} \sin a'OY.$$

Making the substitution $I_s = f a_s'$ we get,

$$\text{Change of epoch at reversal} = \frac{2F}{I_s} \sin a'OY. \quad . \quad . \quad . \quad . \quad . \quad (81)$$

The figure shows that when the maximum positive displacement is being approached, Oa' will be retarded or advanced according as the reversal is from increasing to decreasing displacement or *vice versa*. If the maximum positive displacement has been passed the opposite rule holds.

As the displacement due to the actual synchronizing vibration at any position of Oa' is $Oa' \cos a'OY$ the velocity must be

$$\omega_s \cdot Oa' \sin a'OY.$$

Hence the change of epoch at reversal is proportional to the ordinate of the velocity curve for the actual synchronizing vibration, at which reversal occurs.

Thus in the velocity curve diagram, Fig. 16_1, the velocity curve due to the actual synchronizing term is $a'b'c'$, and $a_1b_1c_1$ is a portion of the reversed velocity curve due to the rest of the terms.

The reversal at d is due to the equal and opposite velocities de and the consequent change of epoch at this reversal is protional to dc.

The numerical value of the change in radians is, equation (81),

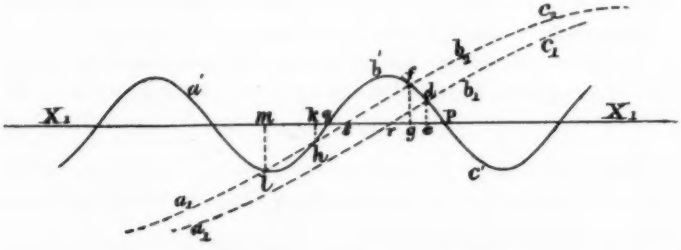
$$\frac{2F}{I_s} dc, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

the amplitude of $a'b'c'$ being reckoned unity. As the reversal is from increasing to decreasing displacement the epoch changes towards 180°. If $a_1b_1c_1$ had passed through p, the 0° ordinate of Fig. 21, would have been that at which rate of energy transfer must be measured, and the change of epoch would have been zero. If it had passed through q, 180° would have been the ordinate for measuring rate of energy transfer, and the change of epoch would again have been zero. The actual ordinate from which rate of energy transfer must be measured is therefore

$$\frac{rp}{pq} \times 180° = \varepsilon_1'.$$

Thus we can plot the curve so long as there is only a single reversal at each end of the valve travel.

If the normal ordinate is changed so that the reversed velocity curve occupies the position $a_2b_2c_2$, we have three reversals f, h, and l. Both at f and l the change is from increasing to decreasing displacement. But q being a point of zero velocity of the



Velocity Curves

Fig. 16.

actual synchronizing vibration, corresponds to Oa', Fig. 16, passing YY_1. Thus as reversal at f retards Oa'—that is, increases ε_1'—reversal at l will accelerate it—diminish ε_1'. The reversal at h must have an opposite effect from that at l; that is, it must retard Oa'—increase ε_1'. To find the total retardation for this

Fig 16.

Curves of Change of Epoch at Reversal, Due to Friction.

The scale gives the total change of epoch, $\frac{\phi}{2}K$ radians, at end of the valve travel. The change is always towards epoch 180°.

The number of each curve gives the ratio of the maximum inertia stress of the synchronizing term to the inertia of the rest of the valve. Motion at the end of the valve travel = $\frac{\phi}{2e}$ or $\frac{1}{2f}$.

A = Normal epoch.
Curve aaaa gives the rate of increase of epoch due to friction per half revolution
Curve A B C gives the rate of restoration of normal epoch due to energy from the gear, per half revolution.
Scale of epoch.
0° 20° 40° 60° 80° 100° 120° 140° 160° 180° 200° 220° 240° 260° 280° 300° 320° 340° 360°
Scale of k
15 1 .5 0 .5 1 15
The scale gives the total change of epoch at one end of the valve travel.
A
B
C
a
a
a
a
n = 200
x
x

Fig 16.

set of reversals we must diminish fg by the difference of lm and hk. Replacing de in equation (82) by $(fg + hk - lm)$ gives the retardation in radians. If the synchronizing vibration is of odd period the effect at the other extremity of the stroke of the valve will be similar and the total effect be doubled for one revolution.

Fig. 16_2 gives a series of these curves of change of epoch at reversal. This figure is constructed like Fig. 21, the numbers of the curves giving the ratio of maximum stress of the actual synchronizing vibration to the inertia of the rest of the valve motion at the end of the travel; that is, the ratio I_s/I_e or I_s/I_f, I_e and I_f being the inertia from the nonsynchronizing terms at opposite ends of the rigid gear valve travel. The scale at the end gives the total change of epoch in radians from the reversal or reversals at one end of the valve travel. It will be noted how very quickly the change of epoch falls off as we pass from the epoch at which triple reversals begin, towards the epoch 180°. Indeed, it can at once be seen from Fig. 16_1 that the tangent at each singular point to that branch of the curve which passes through 180° epoch, is normal to the axis of epochs.

The application of these curves to an actual case, while exceedingly simple, must be reserved till some other questions have been examined. See *Chapter IX.*

41. MAXIMUM CHANGE OF ε_s' BY ONE REVERSAL.

From equation (81), putting $\sin a'OY = 1$, we find the maximum change of epoch is $2F/I_e$ when this angle is small. The following is an exact estimate of the maximum change of epoch, which may be of interest though not necessary to what is to follow.

In Fig. 17 let Oe be the position of the revolving arm when reversal occurs and suppose that the effect of reversal is such as to change Oe to Oh, coinciding with XX, or vice versa. Then eh drawn perpendicular to XX will be equal to $2F/f$. I shall show that the change of epoch is then a maximum. The result of course is the same as that which can readily be got by more cumbrous deduction from the general equations.

Suppose reversal to occur when the arm is at Oe', very near Oe. $Oe' = Oe$. Draw $e'h'l$ perpendicular to XX and make $e'h' = 2F/f$.

Oh' would then be the position of the revolving arm, after reversal. Draw em perpendicular to $e'h'$.

The difference in those two changes of epoch—that is, the difference of the angles eOh and $e'O'h'$ is the same as the difference of the angles hOh' and eOe'.

$$he = lm = e'h' = \frac{2F}{f}.$$

Therefore $$e'm = lh'.$$

The angle $$hOh' = \frac{lh'}{Oh'} = \frac{e'm}{Oh},$$

since ee' and em are extremely small compared with Oe.

By similar triangles Oeh and $e'em$

$$\frac{e'm}{Oh} = \frac{ee'}{Oe} = \text{angle } eOe'.$$

Thus there is no alterations in change of epoch for a very small shift from Oe on either side. If the shift is considerable it will readily be perceived that the consequent change of epoch at reversal is sensibly less. Hence the change is a maximum when from Oe to Oh or vice versa. Its value is

$$\text{Angle } eOh = sin^{-1}\frac{2F}{a_s'f} =$$
$$= sin^{-1}\frac{2F}{I_s}.$$

42. RATE OF RESTORATION OF NORMAL EPOCH BY GROWTH OF a_s'.

Let us now examine the effect which the growth of the actual synchronizing vibration, through energy communicated by the gear, has in restoring normal epoch.

To recapitulate,—cf, Fig. 16, gives the growth of the vector Oc, which vector at each moment determines the displacement and epoch of the actual synchronizing vibration. This growth, cf, is always parallel to the normal vector Oa, and is at the rate $a_s'\omega_s f/2$, equation (62). Since $\omega_s t$ increases by 2π in one cycle of the synchronizing vibration it must increase by $2s\pi$ per revolu-

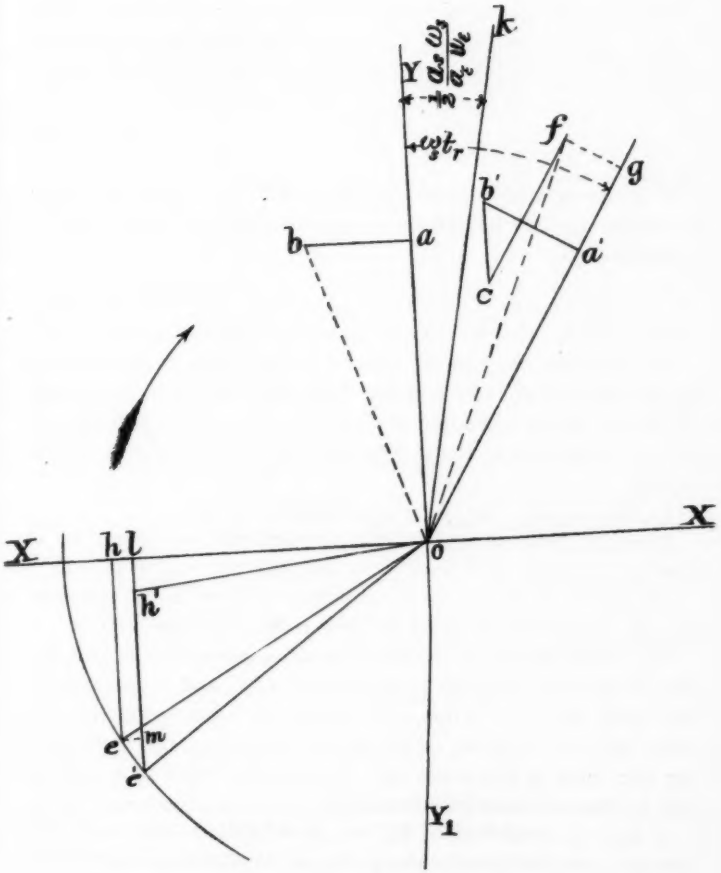

k
Y
$\frac{a_s}{a_c}\frac{\omega_s}{\omega_c}$
$\frac{1}{2}$
$\omega_s t_r$
f
g
b'
b
a
a'
c
X
X
h l
o
h'
e
m
c'
Y_1

Fig 17.

tion of the engine. Hence the whole growth of cf per revolution is,

$$cf = a_s s\pi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (83)$$

The angle aOc, by which the normal epoch is departed from, equals α^*; then, if fO be joined, the angle fOc is approximately,

$$\frac{cf \sin \alpha}{Oc},$$

if cf is comparatively small compared with Oc. And the angle by which the normal epoch is approached per revolution is, equation (83),

$$\frac{a_s s\pi}{Oc} \sin \alpha = \frac{a_s s\pi}{a_s'} \sin \alpha = \frac{fsa_s\pi}{fa_s'} \sin \alpha = \frac{fsa_s\pi}{I_s} \sin \alpha. \quad . \quad (84)$$

To make the effect of the various actions clear let us revert to the simple case of §35 where the rigid gear valve motion is made up of two simple harmonic motions only, one of the same period as the revolution of the engine and one of odd synchronizing period.

In Fig. 16_3 let A be the normal epoch.

Then the sine curve ABC will give the rate of restoration of normal epoch per *half* revolution deduced from equation (84).

The curve *aaaa* . . . is the curve $I_s/I_e = 2.00$ of Fig. 16_2, giving the retardation of epoch per half revolution from friction.

The ordinates above the zero line are supposed to represent a rate of decrease of epoch for the curve ABC, and of increase for the curve *aaa* . . . Thus, at B where the curves cut, the two rates balance. And as, in the simple case taken, the conditions are the same at the other end of the valve travel, the balance will be true over a whole revolution.

If the 2d condition of §38 be also fulfilled—that is, if the energy communicated by the gear and that abstracted by friction balance for epoch B—the elements of the actual synchronizing vibration would continue practically without change. If these

* Oc, Fig. 16, was originally the change from Oa produced by one reversal, but is not now being used in this sense. Oa is still the normal vector, but Oc any actual vector, the angle $aOc = \alpha$ having any magnitude.

energies did not balance the amplitude a_s' would change and new curves *ABC*, *aaa* . . . would have to be drawn for larger or smaller values of a_s' till a balance was obtained and both conditions of §38 complied with.

The curve *ABC* I have drawn for the case of the *Newark's* L.P. gear at .76 cut off, *Table X;* for which, however, J is not very small.

F	=	2,163 lbs.	Line (*aa*), *Table X.*
I_s	=	12,580 "	" (*ee*), " " .
f_s	=	2,948,244 "	" (*q*), " " .
a_s	=	$10^{-3} \times 1.505$ feet.	" (*bb*), " " .
Normal epoch	=	114°	" (*v*) " " .

The change per *half* revolution giving the curve *ABC* is, from equation (84)

$$\frac{fsa_s\pi}{2I_s} \sin a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

The unit of the vertical scale, Fig. 16_3, is—

$$\frac{2F}{I_s} = \frac{4326}{12580} = .3439 \text{ radian per half revolution.}$$

The amplitude of curve *ABC* is

$$\frac{fsa_s\pi}{2I_s} = \frac{2{,}948{,}244 \times 10^{-3} \times 1.505 \times \pi}{2 \times 12{,}580} = .554 \text{ radian.}$$

Hence the amplitude of *ABC* to scale is

$$\frac{.554}{.3439} = 1.611.$$

43. DETERMINATION, AND CURVES, OF ENERGY TRANSFER DUE TO FRICTION, AT REVERSAL.

I will now explain the method of determining the rate of energy transfer due to friction shown in curves, Fig. 21.

Since we are regarding a_s' as constant during each revolution—that is, dealing only with its mean value—we may replace the complete expression for the actual synchronizing vibration in equation (62),

$$\left(\frac{B_0 + a_s\omega_s t}{2}\right) sin\,(\omega_s t - \varepsilon_s) + A\,cos\,(\omega_s t - \lambda),$$

by

$$a_s'\,cos\,(\omega_s t - \varepsilon_s').$$

We can then proceed, as we did from equation (62), and find an equation precisely the same as equation (66) except that a_s' is substituted for A. The amplitude a_s' before reversal is reduced to the square root of

$$a_s'^2 - \frac{4Fa_s'}{f}\,cos\,(\omega_s t_r - \varepsilon_s') + \frac{4F^2}{f^2}.$$

The energy is changed from

$$\tfrac{1}{2} f a_s'^2$$

to

$$\tfrac{1}{2} f a_s'^2 - 2Fa_s'\left\{ cos\,(\omega_s t_r - \varepsilon_s') - \frac{F}{fa_s'}\right\}.$$

By supposition $F/fa_s' = J'$, is small, and we may take the total change of energy as

$$(p) \qquad -2Fa_s'\,cos\,(\omega_s t_r - \varepsilon_s'). \quad . \quad . \quad . \quad . \quad . \quad (86)$$

This is for a reversal from increasing displacement to decreasing, as noted after equation (66). If reversal is from decreasing to increasing displacement the total change of energy becomes

$$(q) \qquad +2Fa_s'\,cos\,(\omega_s t_r - \varepsilon_s'). \quad . \quad . \quad . \quad . \quad . \quad (87)$$

Figs. 18 and 19 will illustrate the application of expressions (86) and (87). They are similar to Figs. 10 and 11. The velocity at O_1 is negative, from Fig. 19. The first intersection between the velocity curve $a'b'c'$ and the reversed velocity curve $a_1b_1c_1$ is at d' where the dotted line q_1 is drawn up to the upper figure. The reversal is from decreasing to increasing displacement as shown by Fig. 19. Thus the change of energy must be reckoned from expression (87). The ordinate of $a'b'c'$, Fig. 18, at this time is

$$de = a_s'\,cos\,(\omega_s t_r - \varepsilon_s').$$

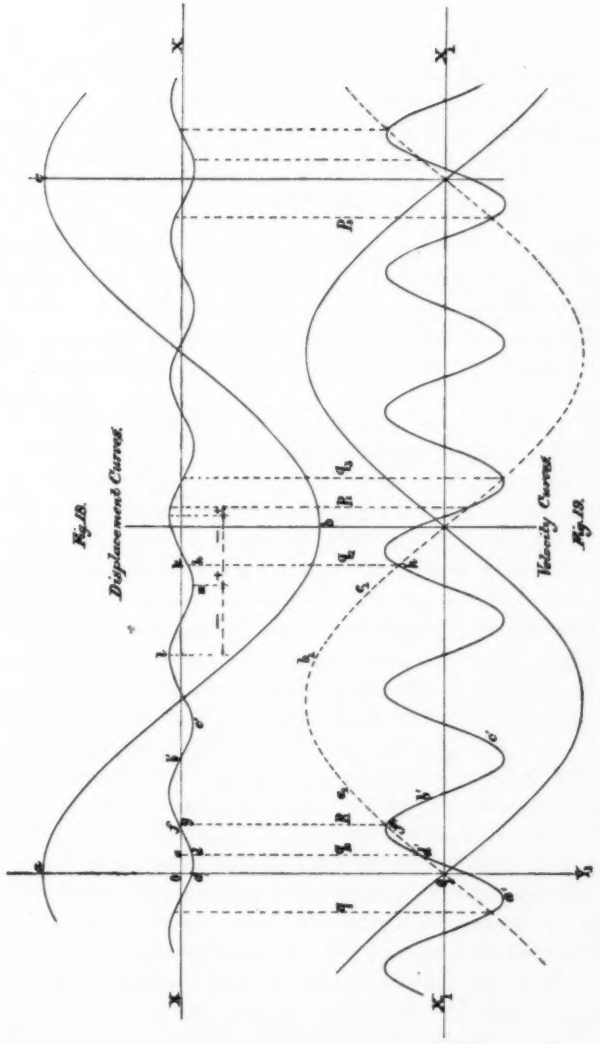
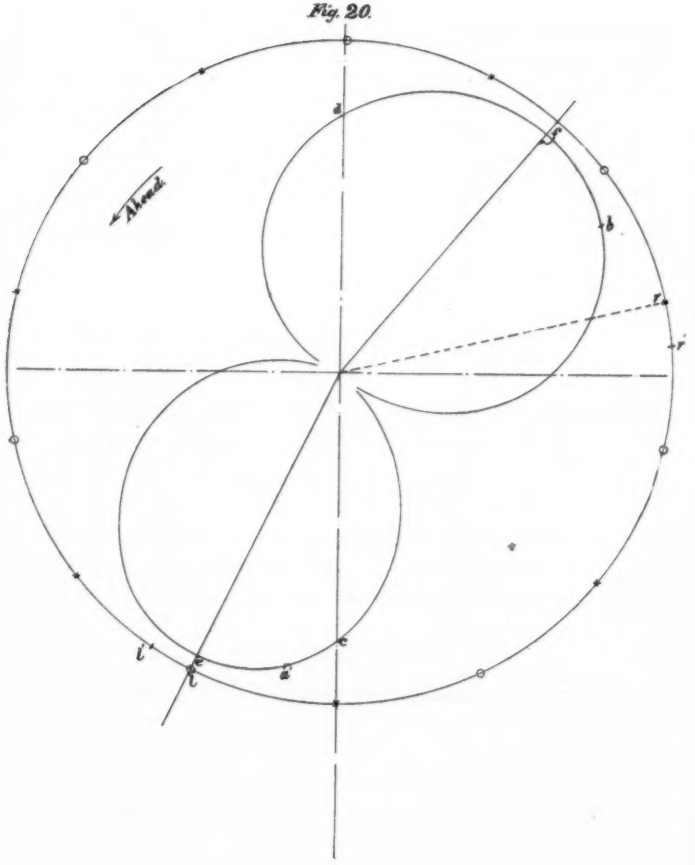


Fig. 13.
Displacement Curves.
Fig. 12.
Velocity Curves

Fig. 20.



This is negative and therefore a quantity of energy

$$2F \times de$$

is to be deducted from the actual synchronizing vibration.

The next reversal is at f' and the ordinate fg is positive. As it takes place from increasing to decreasing displacement expression (86) must be used. Hence a quantity of energy

$$2F \times fg$$

must again be deducted from the actual synchronizing vibration.

At the next reversal, h', the valve is nearing the other end of its stroke. It is indifferent whether we count the reversal from decreasing to increasing displacement and consider hk negative, or from increasing to decreasing displacement and consider hk positive.

The process is readily followed through the whole revolution and the transfer of energy, $\tau F a_s'$, measured. The coefficient of energy transfer due to friction, τ, is given in Figs. 21 and 22 for all epochs of the actual synchronizing vibration and for a series of values of $n_e = I_s/I_e$, as already fully explained in §29.

In the case given in Figs. 18 and 19 energy is abstracted at each reversal. If the curve $a'b'c'$, Fig. 19, moves to the right the points of reversal d' and f' will approach one another. When the points coincide and $a_1b_1c_1$ is tangent to $a'b'c'$ there will be a singular point on the energy transfer curve; as, for still further change of ϵ_s' the energy transfer will follow a new law, the reversal being single.

There is one other way of looking at the question, which is more obvious than that just given. In Fig. 18 between l and m the valve moves downward by an amount due to the sum of the change of ordinate of both curves abc and $a'b'c'$. Energy is lost in friction both by the synchronizing vibration and the rest of the terms which give the curve abc. On passing the maximum downward displacement m of $a'b'c'$ the movement due to the synchronizing vibration is upward, though the total movement of the valve is downward. Friction thus restores energy to this vibration over the change of ordinate from m to h. At

h, which is a point of reversal, the conditions are again reversed and energy is lost. Thus abstraction of energy changes to restoration, or vice versa, at every maximum displacement of the curve $a'b'c'$ and also at every point of reversal; and the energy transferred is equal to F multiplied by the change of displacement of $a'b'c'$. Following this method over the whole revolution leads to the same results as the use of expressions (86) and (87).

As the motion will on the average be quicker when that due to both curves is in the same direction than when they are opposite, and as equation (26) shows that the coefficient of friction falls slightly as the velocity of sliding increases, the average rate of abstraction of energy per unit length will be smaller than the rate of communication. Hence we have here a further cause of growth of the actual synchronizing vibration.

If Fig. 19 were drawn to scale it would be found that until the amplitude of the actual synchronizing vibration was such that at each end of the valve travel there were more than three reversals, the portion of the curve $a_1b_1c_1$ at which reversals occur differs but slightly from a straight line. And, as we have seen in §26, the ratio of the maximum stresses due to the long and short period curves is the same as the ratio of the tangents of the angles with which their velocity curves cut the axis of X_1X_1. Hence, although the curves of Fig. 16_2 are drawn for a synchronizing vibration of the 5th period, they differ but slightly from those of any higher period. To be more explicit,—the portions for single reversals are sensibly the same for all periods; but, as the synchronizing period becomes higher, the singular points are more slightly nearer the 180° epoch. Thus, if the synchronizing period is raised from the 5th to infinity—by substituting the tangent at the zero velocity point for the portion of the curve $a_1b_1c_1$, Fig. 19, at which reversals occur—we will find the epochs of the singular points in Fig. 16_2 not sensibly affected for the lower values of n_e; but as n_e rises this shift increases till, for $n_e = 3.4$, it is about 15°. The same shifts of the singular points necessarily occur in Fig. 21.

44. ENERGY SUPPLIED BY THE GEAR TO THE ACTUAL SYNCHRONIZING VIBRATION, PER REVOLUTION.

The energy supplied to the gear by the actual synchronizing vibration of the valve per cycle of this vibration is given by equation (25_1) as

$$\pi f a_s a_s'$$

when the actual synchronizing vibration has its normal epoch, and the change of a_s' per cycle is a small fraction of its mean value. The meaning of this is clear from Fig. 16. Oa being the normal vector, the force evected on the valve to displace it by a distance Oa is fa_s'. The energy supplied by the gear will increase Oa at the rate

$$\frac{a_s \omega_s t}{2};$$

or, per cycle—see §25—by,

$$a_s \pi.$$

The energy required to overcome the force fa_s' through the distance $a_s\pi$ is

$$\pi f a_s a_s'$$

as given above.

If Oc is the vector of the actual synchronizing vibration the growth of cf is at the angle α to it. Hence, while cf per cycle is $a_s\pi$, the growth of Oc is only

$$a_s \pi \cos \alpha;$$

and the energy supplied by the gear per cycle is

$$\pi f a_s a_s' \cos \alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

(The analytical deduction of this equation is given in Appendix D—See equation (20d).)

Hence,—

Energy from gear per revolution of engine

$$= \pi f a_s a_s' s \cos \alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (89)$$

45. EXPRESSION OF 2D CONDITION, §38.

As the events of one revolution are repeated in the next only when the energy supplied by the gear balances that abstracted

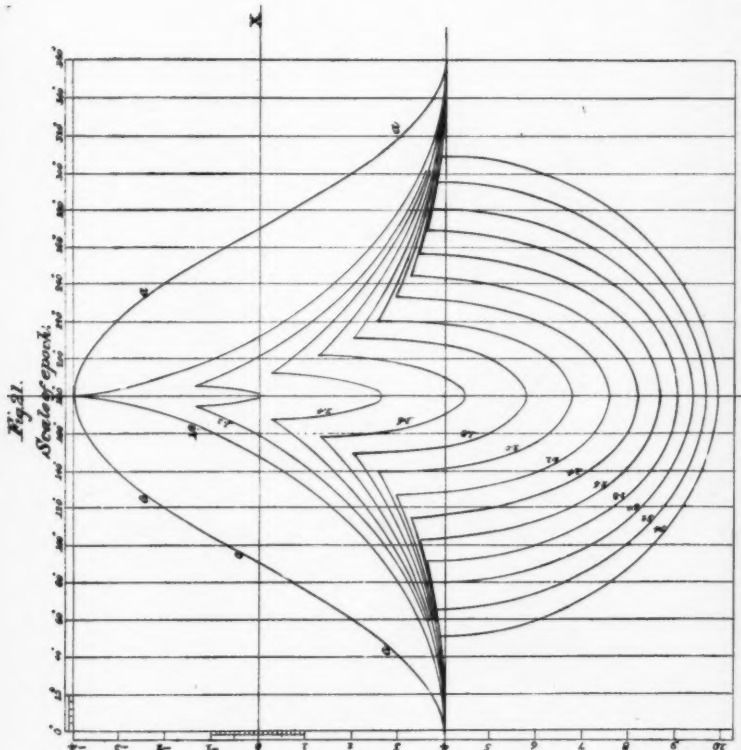


Fig. 21.

Scale of epoch.

Curves of Energy Transfer Due to Friction.

The number of each curve gives the ratio of the maximum inertia stress of the advancing term to the inertia of the rest of the entire motion at the end of the stroke. I_2/I or I_2/I_1. Transfer of energy per revolution $= \pi F a$.

Curves of Energy Transfer Due to Friction.

TABLE VI.

HARMONIC ANALYSIS OF HORIZONTAL MOVI

$\frac{k}{R}$	$\frac{a}{R}$	Q_0	$\frac{Q_1}{Q_0}$	Q_1	$\frac{Q_2}{Q_1}$	Q_2	$\frac{Q_3}{Q_2}$	Q_3	$\frac{Q_4}{Q_3}$	Q_4	$\frac{Q_5}{Q_4}$
.8	.12	$10^{-1}\times 4.664$	1.179	$10^{-1}\times 5.497$	.2629	$10^{-1}\times 1.445$	.2789	$10^{-2}\times 4.03$	.2928	$10^{-2}\times 1.18$	.30
	.1	$10^{-1}\times 5.145$	.8270	$10^{-1}\times 4.255$	.2115	$10^{-2}\times 9.00$	.2245	$10^{-2}\times 2.02$	.235	$10^{-3}\times 4.74$	.24
	.08	$10^{-1}\times 5.485$	.588	$10^{-1}\times 3.226$	.165	$10^{-2}\times 5.32$	.175	$10^{-3}\times 9.3$	.183	$10^{-3}\times 1.70$	.18
	.06	$10^{-1}\times 5.722$	.407	$10^{-1}\times 2.327$	.121	$10^{-2}\times 2.825$	.128	$10^{-2}\times 3.62$	.135	$10^{-4}\times 4.9$	.13
.7	.188	$10^{-1}\times 5.918$	.780	$10^{-1}\times 4.618$	.289	$10^{-1}\times 1.333$	.293	$10^{-2}\times 3.908$	.310	$10^{-2}\times 1.212$	.31
	.17	$10^{-1}\times 6.196$	.639	$10^{-1}\times 3.962$	.255	$10^{-1}\times 1.011$	.258	$10^{-2}\times 2.61$	.274	$10^{-3}\times 7.146$	.28
	.12	$10^{-1}\times 6.720$	.375	$10^{-1}\times 2.519$	.173	$10^{-2}\times 4.348$	.173	$10^{-3}\times 7.53$	.184	$10^{-3}\times 1.384$	.18
	.08	$10^{-1}\times 6.964$	.229	$10^{-1}\times 1.597$	.113	$10^{-2}\times 1.799$	.1125	$10^{-3}\times 2.024$	.120	$10^{-4}\times 2.42$	.12
.6	.24	$10^{-1}\times 7.047$	.503	$10^{-1}\times 3.546$	.290	$10^{-1}\times 1.029$	.273	$10^{-2}\times 2.81$	.295	$10^{-3}\times 8.3$	.30
	.2	$10^{-1}\times 7.389$	.369	$10^{-1}\times 2.728$	.235	$10^{-2}\times 6.422$	.219	$10^{-2}\times 1.406$	.237	$10^{-3}\times 3.334$	.24
	.15	$10^{-1}\times 7.680$	.248	$10^{-1}\times 1.906$	.173	$10^{-2}\times 3.29$	.158	$10^{-3}\times 5.21$	.173	$10^{-4}\times 9.0$	.17
	.1	$10^{-1}\times 7.864$	.154	$10^{-1}\times 1.214$	.113	$10^{-2}\times 1.374$	.103	$10^{-3}\times 1.418$	.115	$10^{-4}\times 1.627$	.11
.42	.34	$10^{-1}\times 8.292$	.296	$10^{-1}\times 2.451$	.343	$10^{-2}\times 8.398$	.246	$10^{-2}\times 2.064$	.297	$10^{-3}\times 6.139$	.28
	.28	$10^{-1}\times 8.583$	.215	$10^{-1}\times 1.847$	.279	$10^{-2}\times 5.147$	.193	$10^{-3}\times 9.94$	.238	$10^{-3}\times 2.363$	.22
	.2	$10^{-1}\times 8.841$	.137	$10^{-1}\times 1.215$	.197	$10^{-2}\times 2.395$	.132	$10^{-3}\times 3.16$	.165	$10^{-4}\times 5.22$	.15
	.13	$10^{-1}\times 8.980$	.084	$10^{-2}\times 7.541$	.1275	$10^{-3}\times 9.617$	.0836	$10^{-4}\times 8.04$	.106	$10^{-5}\times 8.51$	.10
$\frac{1}{3}$	.4	$10^{-1}\times 8.615$	.246	$10^{-1}\times 2.117$	.412	$10^{-2}\times 8.73$	.233	$10^{-2}\times 2.03$	.323	$10^{-3}\times 6.56$	.28
	.34	$10^{-1}\times 8.879$	.185	$10^{-1}\times 1.646$	.351	$10^{-2}\times 5.784$	.188	$10^{-2}\times 1.085$	.269	$10^{-3}\times 2.914$	.23
	.24	$10^{-1}\times 9.175$	.114	$10^{-1}\times 1.047$	.247	$10^{-2}\times 2.59$	.1255	$10^{-3}\times 3.25$	.185	$10^{-4}\times 6.0$	.15
	.15	$10^{-1}\times 9.333$	.066	$10^{-2}\times 6.18$	.1545	$10^{-3}\times 9.55$	.0756	$10^{-4}\times 7.22$	.1143	$10^{-5}\times 8.25$	.09

BLE VI.

ONTAL MOVEMENT OF A, FIGS. 4 TO 7.

Q_4	Q_5/Q_4	Q_5	Q_6/Q_5	Q_6	Q_7/Q_6	Q_7	j	Q'_0	Q'_1
$^{-2}\times 1.18$	.3025	$10^{-3}\times 3.57$	.3100	$10^{-2}\times 1.11$	.3135	$10^{-4}\times 3.48$	.315	1.131	.3562
$^{-3}\times 4.74$	.243	$10^{-3}\times 1.15$	.247	$10^{-4}\times 2.84$	.250	$10^{-5}\times 7.1$	.252	1.100	.2772
$^{-3}\times 1.70$	.188	$10^{-4}\times 3.2$	.191	$10^{-9}\times 6.12$	.1945	$10^{-5}\times 1.19$	.196	1.071	.2099
$^{-4}\times 4.9$	.138	$10^{-5}\times 6.76$	.141	$10^{-6}\times 9.52$	.143	$10^{-6}\times 1.36$	.144	1.059	.1525
$^{-2}\times 1.212$	.3175	$10^{-3}\times 3.848$	.323	$10^{-3}\times 1.243$	.327	$10^{-4}\times 4.07$	.329	.9754	.3209
$^{-3}\times 7.146$	.280	$10^{-3}\times 2.002$	.285	$10^{-4}\times 5\,709$	.289	$10^{-4}\times 1\,648$	.290	.9553	.2771
$^{-3}\times 1.384$	.188	$10^{-4}\times 2\,604$	.191	$10^{-5}\times 4.982$	.194	$10^{-6}\times 9.68$	195	.9030	.1761
$^{-4}\times 2.42$	.122	$10^{-5}\times 2.96$	.125	$10^{-6}\times 3.69$	.126	$10^{-7}\times 4.66$	.127	.8746	.1108
$^{-3}\times 8.3$	.300	$10^{-3}\times 2.49$	.303	$10^{-4}\times 7.55$	.306	$10^{-4}\times 2.31$	.306	.9195	.2814
$^{-3}\times 3.334$	.241	$10^{-4}\times 8.033$	.245	$10^{-4}\times 1.97$	.247	$10^{-5}\times 4.87$	.247	.8682	.2145
$^{-4}\times 9.0$	.174	$10^{-4}\times 1.57$	.178	$10^{-5}\times 2.8$	.180	$10^{-6}\times 5.04$	.180	.8233	.1482
$^{-4}\times 1.627$	.112	$10^{-5}\times 1\,826$	.117	$10^{-6}\times 2\,129$	.118	$10^{-7}\times 2.51$	.118	.7880	.0930
$^{-3}\times 6.139$	.288	$10^{-3}\times 1.768$	.299	$10^{-4}\times 5.29$	.300	$10^{-4}\times 1.588$	.300	.7261	.2178
$^{-3}\times 2.363$	.229	$10^{-4}\times 5.40$	.238	$10^{-4}\times 1.287$	.2385	$10^{-5}\times 3\,07$	.2385	.6994	.1668
$^{-4}\times 5.22$	.1575	$10^{-5}\times 8.22$	.165	$10^{-5}\times 1.356$	.165	$10^{-6}\times 2.24$	.165	.6728	.1110
$^{-5}\times 8.51$	.100	$10^{-6}\times 8.54$	.105	$10^{-7}\times 8\,99$	.1052	$10^{-8}\times 9\,46$	.1052	.6634	.0698
$^{-3}\times 6.56$	.288	$10^{-3}\times 1.89$	.3137	$10^{-4}\times 5.93$	.302	$10^{-4}\times 1.79$	.3078	.6838	.2056
$^{-3}\times 2.914$	.236	$10^{-4}\times 6.869$	.258	$10^{-4}\times 1.773$	.2504	$10^{-5}\times 4.44$	.2542	.6473	.1615
$^{-4}\times 6.0$	.158	$10^{-5}\times 9.5$	.1779	$10^{-5}\times 1.69$	.1698	$10^{-6}\times 2.87$	.1738	.5991	.0999
$^{-5}\times 8.25$	.0967	$10^{-6}\times 7.98$	.1084	$10^{-7}\times 8.65$	.1042	$10^{-8}\times 9.01$	.1063	.5875	.0587

by friction, we obtain a most important relation by equating expression (89) to $\tau Fa_s'$, thus,—

$$\pi f a_s a_s' s \cos a = \tau F a_s',$$

or,

$$F = \frac{\pi f s a_s}{\tau} \cos a. \quad . \quad . \quad . \quad . \quad . \quad . \quad (90)$$

This is the expression of the 2d condition of §38.

CHAPTER IX.

CURVES OF ENERGY TRANSFER AND CHANGE OF EPOCH APPLIED.

46. ACTUAL CONDITIONS—DIFFICULTY OF SOME CASES.

The curves in Figs. 16_2 and 21 have been obtained on the assumption that the curve *abc*, Fig. 18, gives the same stresses at *a*, *b*, and *c*, and also that the time intervals from *a* to *b* and *b* to *c* are equal. Neither of these conditions is fulfilled in an actual gear, and Fig. 1 shows that for the *Newark* neither condition is even approximated to. Hence the energy transfer curves cannot be immediately applied to any case just as they stand. But pairs of curves, Fig. 16_2, can always be combined to give the required curve.

Only in the case of gears, for which the size and epoch of the rigid gear synchronizing term can be calculated, is it possible to apply them to predict very definitely the intensity of the inertia stresses that will be induced by the action of the gear. But they throw great light on the nature of the action of all gears and help us to arrive at the general conditions which must be fulfilled to produce a satisfactory design.

For such gears as Stephenson's, Gooch's, &c., the elements of the rigid gear synchronizing term can be calculated with sufficient approximation from the general equations from which equations (10) and (11) have been derived. But in these the amplitude of successive periods diminish so rapidly and the natural period is usually so high that a very small rate of absorption of energy will check the growth of the actual synchronizing vibration. We must recollect, however, that τ may be negative till I_s has become important.

But for many of the more recent gears the calculation of the elements of the rigid gear synchronizing term would be extremely laborious, and difficult to tabulate. For Marshall's, and the practically identical gears which sometimes go under a different name, the calculations of the synchronizing term is possible and the results can readily be shown by a pair of diagrams.

47. DATA REQUIRED, s, I_e AND I_f.

To draw the energy transfer curves for any particular valve gear it is necessary first to know the natural period. This can readily be calculated from the drawings of any gear as shown for the *Newark* in Table VIII and § 64.

Suppose Fig. 29 to be the polar valve diagram. Determine the points e and f for which the diagram shows the greatest displacements of valve. These are the points near which reversals must occur.

Determine the stresses at e and f by equation (6), or a modification given in Chapter II. It will be sufficient to suppose that, omitting the actual synchronizing vibration, the actual stresses are very near those so determined and that they bear the same ratio. Call these stresses I_e and I_f.

The number of the energy transfer curve, Fig. 21, which would have to be used if we had only reversals at e to consider would be given by

$$\frac{I_e}{I_e} = n_e.$$

If reversals at f only had to be considered we would similarly have for n_f, the number of the energy transfer curve to be used,

$$\frac{I_f}{I_f} = n_f.$$

Hence

$$\frac{n_f}{n_e} = \frac{I_e}{I_f}. \quad \ldots \ldots \ldots \quad (91)$$

In the *Newark* for full gear ahead

$$\frac{I_e}{I_f} = 1.65 = \frac{n_f}{n_e},$$

or

$$n_f = 1.65 n_e;$$

so that if we suppose an actual synchronizing vibration giving twice the stress at e we have

$$n_e = 2.0 \text{ and } n_f = 2.0 \times 1.65 = 3.3.$$

The scale in Fig. 21 supposes two similar sets of reversals at the two ends of the valve travel, whereas now we have at e one set and at f one set. The ordinates of the combined curves are thus the mean of corresponding ordinates of the n_e and n_f curves. It thus only remains to find the relative position in which the zero ordinates of these two curves must be placed when combining them. To do this—

48. METHOD OF OBTAINING ENERGY TRANSFER CURVE DUE TO FRICTION.

Draw a circle outside the valve diagram and divide it into s equal parts commencing at l on the radius Oe, Fig. 20. These points are marked by small circles, and it is supposed that

$$s = 7.$$

(For the *Newark*, Fig. 29, we have $s = 6$; thus, choosing $s = 7$ here gives an example of both odd and even natural period.)

Suppose Oe gives a valve displacement to the left in a horizontal engine. If the actual synchronizing vibration had its displacement also a maximum to the left when the crank was at Oe, these points indicated by small circles would show the position of the crank each time the actual synchronizing vibration was at its left maximum displacement.

The coincidence of these left maximum displacements requires that we measure the transfer of energy at e from the zero epoch ordinate of the curve, number n_e, Fig. 21.

Bisect the arcs between the small circles. These bisections are marked by crosses. If the small circles correspond to left maximum diplacements of the actual synchronizing vibration the crosses correspond to right maximum displacements.

Proceeding round the circle in the same direction as the crank let r be the last cross before reaching Of, the maximum right displacement from the valve diagram. Coincidence of Or and Of would require the measurement of the energy transfer at f

from the zero epoch ordinate of the curve number n_f, Fig. 21. But as the arm Oa, Fig. 14, revolves s times as fast as the crank, the epoch ordinate of the n_f curve actually used will be $s \times rOf$ degrees. That is, the $O°$ ordinate of the n_e curve and the $s \times rOf°$ ordinate of the n_f curve must be superposed.

If the epoch of the actual synchronizing vibration, ϵ_s', changed from that supposed, so that the left maximum l moved back to l', the right maximum would also move back, over an equal arc, to r'. The epoch ordinate of the n_e curve giving the energy transfer at e would be $s \times l'Oe$ degrees. That of the n_f curve, giving energy transfer at f, would be

$$s \times r'Of = s(r'Or + rOf) = s(l'Oe + rOf) \text{ degrees.}$$

Thus the $O°$ epoch ordinate of the n_e curve and the $s \times r'Of°$ epoch ordinate of the n_f curve would have to be superposed as before, and the combined curve is unaltered. The energy transfer curve for the engine would be the mean of those two superposed curves. The ordinate of this mean curve to be taken in measuring energy transfer due to friction is $s \times l'Oe°$ from the $O°$ epoch ordinate of the n_e curve.

49. EXAMPLE FROM *NEWARK*.

Fig. 29 shows the full ahead diagram of the *Newark's* L.P. gear and Fig. 30 her energy transfer curves.

These will be fully referred to in connection with this gear but will serve here as a practical illustration of the above process.

The natural period of the *Newark's* L.P. valve gear, is as noted above, the 6th. Hence if the diagram were marked off as in Fig. 20 with small circles and crosses the arc between consecutive circles would be $360°/6 = 60°$; and the arc between contiguous circles and crosses, 30°. If we began spacing the small circles from the radius Oe, as in Fig. 20, we would have—

Angle $eOr = 180° - 30° = 150°$.

The angle eOf is $180° - 14° = 166°$.

Consequently rOf would be $= 166° - 150° = 16°$.

As noted above $n_e = 2.0$ gives $n_f = 3.3$, and the 0° epoch ordinate of the 2.0 curve, Fig. 21, must be superposed on the

$16° \times 6 = 96°$ epoch ordinate of the 3.3 curve, and the mean curve drawn. The 2.0 and 3.3 curves are shown dotted in Fig. 30 and the mean curves in full lines marked *aaaa*. The ordinate at which the energy transfer is to be measured will be the subject of further investigation.

50. METHOD OF OBTAINING CURVE OF CHANGE OF EPOCH DUE TO FRICTION.

A similar combination of curves from Fig. 16_2 will give, for any case, the total change of epoch per revolution, due to friction. In Fig. 30_1 we have the case for the *Newark* at .76 cut off, the two dotted lines being those for $n_e = 2.00$ and $n_f = 3.3$. As in Fig. 30 the 96° ordinate of the n_f curve is superposed on the $0°$ ordinate of the n_e curve. As at each ordinate the n_e curve gives the change of epoch for the reversal or set of reversals which take place at one end of the valve travel, and the n_f curve the change of epoch at the other end, the full line *aa*—part of which only is drawn—gives the total change per revolution of the engine.

If *AA* is the normal ordinate we see at once that friction will cause the normal epoch to be increased; that is, the ordinates of *aaaa*, Fig. 30, and *aa*, Fig. 30_1, at which transfers of energy and changes of epoch will have to be measured is to the right of *AA*.

51. CURVE OF RESTORATION OF NORMAL EPOCH, AND EXAMPLE FROM *NEWARK*.

We have seen in § 42 that the rate of restoration of normal epoch due to energy communicated by the gear is given by a sine curve, *EFGHK*, Fig. 30_1, passing through zero at the normal ordinate. *ABC*, Fig. 16_3, gives the rate per half revolution for the same case as Fig. 30_1; hence the amplitude of *EFGHK* is double that of *ABC*; or, equation (84),

$$\text{Amplitude } EFGHK = \frac{\pi f s a_s}{I_s} \text{ radians.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (91_1)$$

The amplitude, measured on the left-hand scale of Fig. 30_1, will also be double that of *ABC*; or

$$= 3.222.$$

As this is so large only a portion of *EFGHK* is shown.

At the point F where this curve and aa intersect, the total change of epoch per revolution will be zero, if all the quantities have been properly determined. This of course involves that for the epoch F the total energy transfer per revolution of the engine should also be zero.

Figs. 30_2, 30_3, 30_4, give similar curves for the earlier cut-offs of the *Newark*.

52. EXPRESSION OF 1ST CONDITION, §38.

We may now express the 1st condition of §38.

In Fig. 30_1 let the left-hand scale reading at F of the ordinate of aa be κ; that is, the change of epoch per revolution due to friction is

$$\frac{2F\kappa}{I_s} \text{ radian.}$$

Equating this to the restoration of normal epoch per revolution, equation (84), since the total change must be zero, we have

$$F = \frac{\pi f s a_s}{2\kappa} \sin a. \quad . \quad . \quad . \quad . \quad . \quad . \quad (91_2)$$

Since this must be the same value of F as is given by equation (90) we have the further relation

$$\tan a = \frac{2\kappa}{\tau} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (91_3)$$

CHAPTER X.

HARMONIC ANALYSIS OF MARSHALL GEAR.

53. SIMPLE APPROXIMATE LAW OF HIGHER PERIOD AMPLITUDES, EQUATIONS (12) TO (15).

Soon after equations (12) to (15) had been calculated, I noticed that, as the series proceeds, the numerical coefficients, or amplitudes approached more and more nearly to a geometrical progression. Whether this expresses exactly the ultimate law of amplitude or not would probably be somewhat difficult to establish and is not of great practical importance. For, as shown in Table III, these coefficients, when of high period, are so sensitive

to changes in the mean angle COX, Figs. 4 to 7, of the radius rod OA, and in the extent of the vertical movement of A, that it is only safe in practice to keep the amplitude of the rigid gear synchronizing term well within the limit which could give trouble. Hence a considerable percentage of error, especially for these high periods, matters little.

Fig. 23 will, I think, leave little doubt that the error in assuming that the higher period amplitudes follow a geometrical series is slight, at least for a considerable number of terms after that giving the 7th period.

We shall see immediately that the reduction of any Marshall gear to an equivalent simple gear, such as is shown in Figs. 4 to 7, so far as the treatment of the higher periods is concerned, could very readily be effected. Hence what is required is to calculate a considerable number of equations such as (12) to (15) extending over a sufficient range of values of k/R and a/R, and then interpolate intermediate values.

54. ADDITIONAL RESULTS LIKE EQUATIONS (12) TO (15).

The result of the calculations is shown in *Table VI.* The equation to the vertical motion of A, Fig. 4, being $y = k + a \cos \theta$, that to the horizontal motion is

$$\frac{r}{R} = Q_0 - Q_1 \cos \theta - Q_2 \cos 2\theta - Q_3 \cos 3\theta - \ldots \quad (92)$$

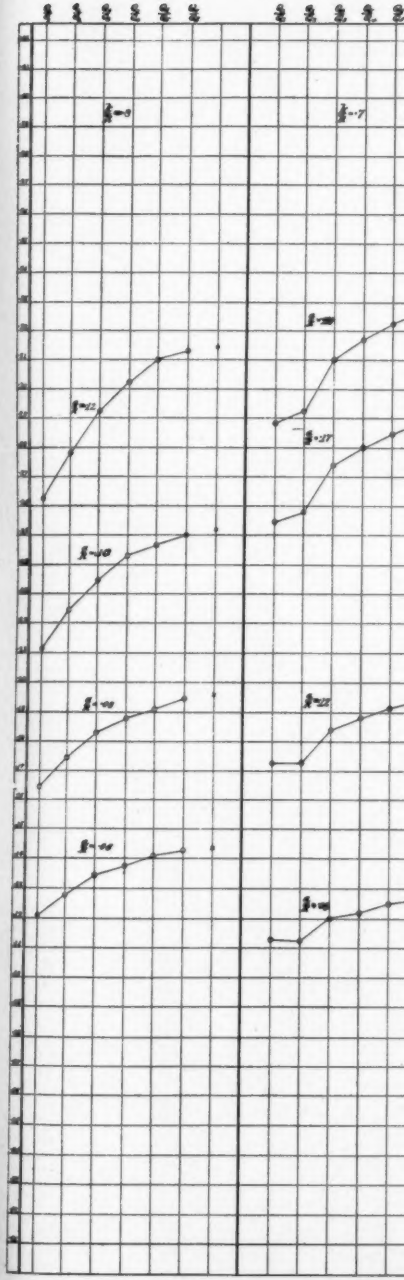
In the first two columns are given the values of k/R and a/R. In eight subsequent columns, on every second line, are given the values of Q_0, Q_1, Q_2, &c. On intermediate lines and not directly under the headings Q_0, Q_1, Q_2, &c., are given the ratios Q_1/Q_0, Q_2/Q_1, &c., from the values in the lines above. These ratios, beginning with Q_2/Q_1, are plotted in Fig. 23, which will be readily understood.

The values of Q up to Q_5 are probably correct to the last decimal. Q_6 and Q_7 possibly contain one or two per cent. of error.*

* The regularity of the change in passing from one set of ratios to the next, clearly exhibited in Fig. 23, and also the numerous checks which were applied as the calculation advanced, while giving the assurance that no large blunder has crept in, do not preclude the possibility of the small errors noted, as some comparatively unimportant

The number of each curve gives the ratio of the maximum circular stress of the synchronizing term to the principal of the rest of the value measured at the end of the extension $\frac{1}{2}/\frac{1}{2}$ or $\frac{1}{2}/\frac{1}{2}$. Transfer of energy per revolution $= \pi^2 a^4$

XL



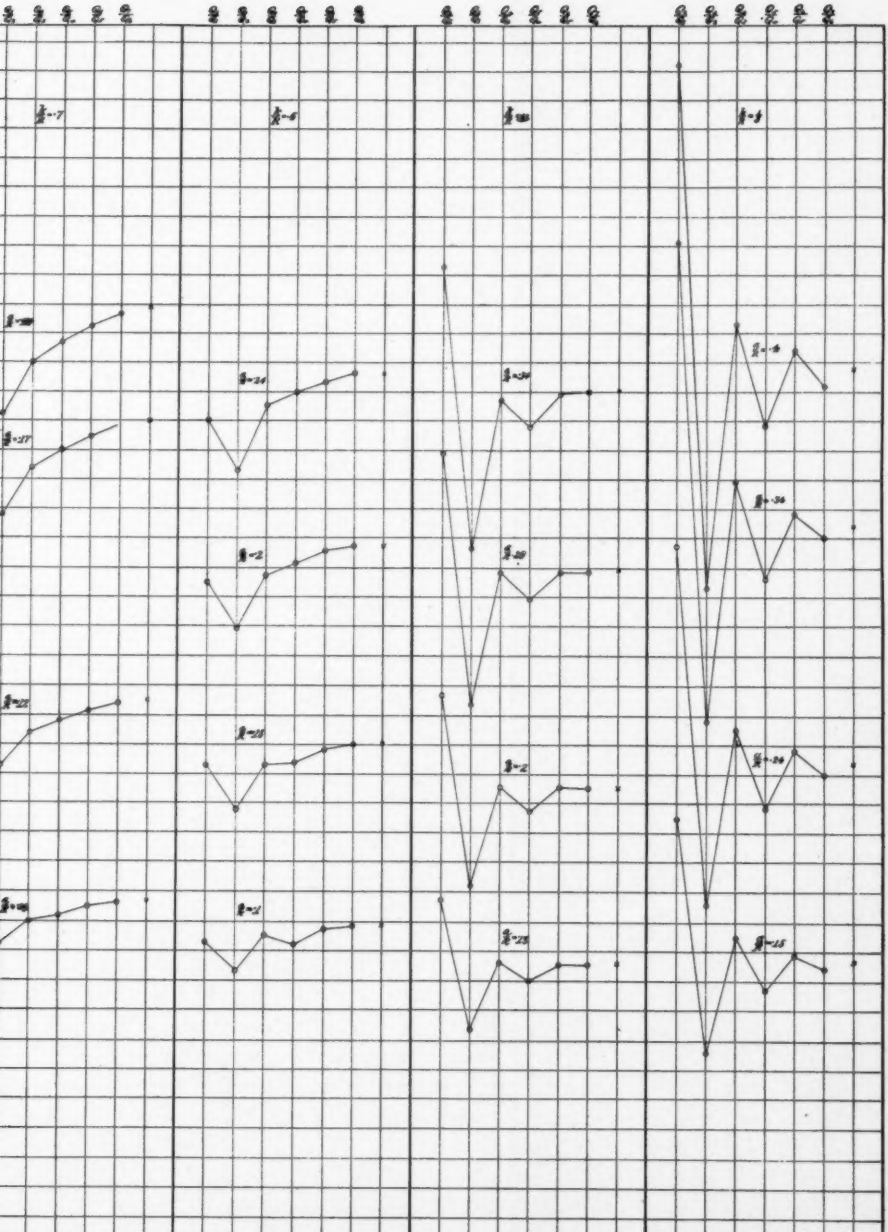


FIG. 23.

55. SUBSTITUTED GEOMETRICAL SERIES.

The third last column of Table VI gives the ratios taken for the substituted geometrical series. These are shown by crosses in Fig. 23. There is a slight margin of choice in fixing on these ratios, but not such as is in any case practically important. For $k/R = 1/3$, the plotted ratios seem to suggest rather higher values than those chosen. But this so distorted the interpolation curves, used in finding intermediate values for Fig. 24, that the values were lowered to those shown. Calling the chosen ratio j, a new first term, Q_0', was calculated from

$$Q_0' = Q_7/j^7.$$

The values of Q_0' from the second last column of *Table VI.*

To calculate the amplitude of the terms of nth period we have

$$Q_n' = Q_0' \times j^n. \quad . \quad . \quad . \quad . \quad . \quad . \quad (93)$$

Figs. 24 and 25 show the tabulated values of Q_0' and j, their position in the diagram being shown by circles. The abscissae give values of a/R_1, R_1 being the length of the radius rod OA, Fig. 4, in the equivalent simple gear, which we shall determine immediately. The values of k/R_1 are marked at the ends of the curves. The ordinates of Fig. 24 give the values of Q_0'/R_1 and those of Fig 25 the values of j/R_1.

The curves not calculated were interpolated graphically.

A very slight percentage of change in the chosen value of j affects by a much larger percentage the derived value of Q_0', and it is also affected by the same percentage of error as exists in Q_7. Hence, without great additional labor, we could not expect Fig. 24 to be free from sensible error. The interpolation curves suggested that there is an error, with a maximum value of somewhere about 2 per cent., which it reaches for $a/R_1 = 2.5$ and $k/R_1 = .60$. These unimportant errors are left as their correction, otherwise than by calculation, would merely be guess work.

---

remainders in the N and P series (see Appendix C), which would have involved great labor to calculate, were estimated; no doubt very approximately.

In the increasing zig-zag of the lines as k/R diminishes we see evidence of the dying out of the odd period vibrations which have completely disappeared in equation (14) where $k/R = 0$.

The form of the interpolation curves between $k/R_1 = .7$ and $k/R_1 = .8$ are also slightly indefinite, but to an entirely unimportant degree.

In Fig. 25, for a constant value of j/R_1, the changes in k/R_1 are nearly proportional to the change in a/R_1, and are such that if the divisions were continued up to the ordinate $a/R_1 = 0$, k/R_1 would reach the value 1.00. So very nearly was this law fulfilled that its possible truth was strongly suggested. I did not, however, use it as the alteration necessary in the values of j would have been a matter of choice and the consequent alterations in Fig. 24 would have been comparatively large.

56. DIVISION OF TOTAL VALVE MOVEMENT.

Let us now analyze the motion given to the valve by the Marshall gear.

Fig. 26 is a diagram of the gear in which the parts are, for clearness, out of practical proportions. The point of suspension, A, of the eccentric beam is between the shaft and the point A_1 at which the radius rod, A_1V, to the valve stem is attached. This is as in the *Newark's* gear, but the name "Marshall Gear" is more usually applied to the modification in which A is outside A_1. The analysis is the same for both forms.

Take XX_1 as the horizontal center line and YY_1 as the vertical center line through the shaft s. E is the center of the eccentric, ES making the angle θ with SY. O is the outer end of the reversing lever from which the radius rod OA is suspended. As usually arranged, when the extremity A of OA is on YY_1, E is on XX_1; the reversing shaft is also then put at A and the lead of the valve is constant.

If we now suppose the center of the eccentric beam E brought to S, A will rise very slightly above the position shown. Let OA then make with the direction XX_1 the angle ψ. Actually, when the lead is nearly constant it will be sufficient to take instead of ψ as defined above, simply the rather larger angle of the reversing lever. If the lead is made variable by increasing ψ above the angle of the reversing lever, its actual value should be taken.

I shall suppose that when the center E is brought to S, the

center of the eccentric beam lies along YY_1. This will lead to sensibly the same result as the exact supposition.

Let

$$ES = a,\ OA = R, \text{ and the ratio } A_1S/AS = m.$$

When the beam center moves from S to E, the radius rod OA will move to OC, and the outer end of the beam from A_1 to C_2. From C as center describe the arc of a circle EF_1, and draw EF perpendicular to YY_1. The eccentric beam may be supposed to move from SAA_1 to ECC_2 by three distinct movements.

(a) The center may shift from S to F, the beam moving to FBB_1.

(b) The center may move to F_1, when the beam will take the position F_1CC_1.

(c) A rotation of the beam about C will bring it to the position ECC_2. These three movements we will consider separately.

(a) *Effect of movement with F.*

57. EQUIVALENT SIMPLE GEAR.

If the lower center of the eccentric beam moved along with F, the point guided by the radius rod OA would travel in an arc of a circle ABC, and the upper center in a curve $A_1B_1C_1$ which, for practical dimensions, is an exceedingly close approximation to an arc of a circle. Let $O_1A_1 = R_1$ be the radius of curvature at A_1, and let $O_1A_1X = \phi_1$. The whole beam has also an angular vibration much smaller in extent than the actual total angular motion due to the eccentric E. For any practical proportion of parts the effect of this angular motion could be left out altogether, but I shall include it with the motion FF_1. Thus for the present

$$\cos BFS = 1.$$

Take x, y, and x_1, y_1, as the coordinates of B and B_1, respectively, parallel to XX_1 and YY_1. For the movement of B take A, and for that of B_1 take A_1, as origin, measuring y downward from those points.

$$x_1 = mx,$$

$$y_1 = y = SF = a \cos \theta.$$

Whence $$\frac{dx_1}{dy_1} = \frac{dx_1}{dx} \cdot \frac{dx}{dy} \cdot \frac{dy}{dy_1} = m\frac{dx}{dy},$$

which gives $$\tan\psi_1 = m\tan\psi\,; \quad \ldots\ldots\ldots \quad (94)$$

also $$\frac{d^2x_1}{dy_1^2} = m\frac{d^2x}{dy^2}.$$

$$\frac{1}{R} = \frac{\dfrac{d^2x}{dy^2}}{\left\{1 + \left(\dfrac{dx}{dy}\right)^2\right\}^{\frac{3}{2}}} = \frac{\dfrac{d^2x}{dy^2}}{\{1 + \tan^2\psi\}^{\frac{3}{2}}} = \frac{d^2x}{dy^2}\cos^3\psi.$$

Similarly $$\frac{1}{R_1} = \frac{d^2x_1}{dy_1^2}\cos^3\psi_1.$$

Hence $$\frac{R_1}{R} = \frac{\dfrac{d^2x}{dy^2}\cos^3\psi}{\dfrac{d^2x_1}{dy_1^2}\cos^3\psi_1} = \frac{1}{m}\frac{\cos^3\psi}{\cos^3\psi_1}.$$

That is $$R_1 = \frac{R}{m}\frac{\cos^3\psi}{\cos^3\psi_1}. \quad \ldots\ldots\ldots \quad (95)$$

Equations (94) and (95) determine the radius O_1A_1 and the angle ψ_1 for the equivalent simple gear.

58. PARTS OF MOVEMENT UNIMPORTANT FOR PRESENT INVESTIGATION.

(b) *Effect of movement FF_1.*

For any gear the varying length FF_1 may be readily determined from a diagram of the gear. If plotted on a time base, supposing θ proportional to the time, it will be found to consist of a nearly pure simple harmonic motion of the 2d period, of small amplitude. The vertical distance of B_1 below O_1 being expressed by

$$y = k + a\cos\theta, \quad \ldots\ldots\ldots \quad (96)$$

it will be found that that of c_1 is very approximately

$$y = k + b + a\cos\theta - b\sin 2(\theta + \varepsilon) \quad \ldots \quad (97)$$

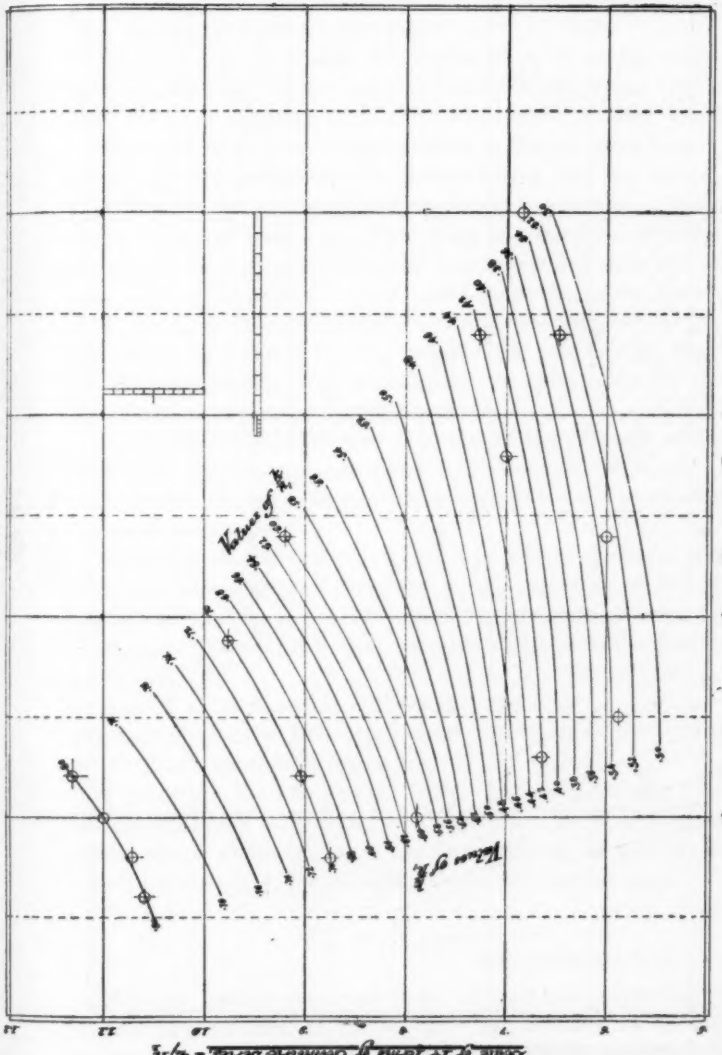
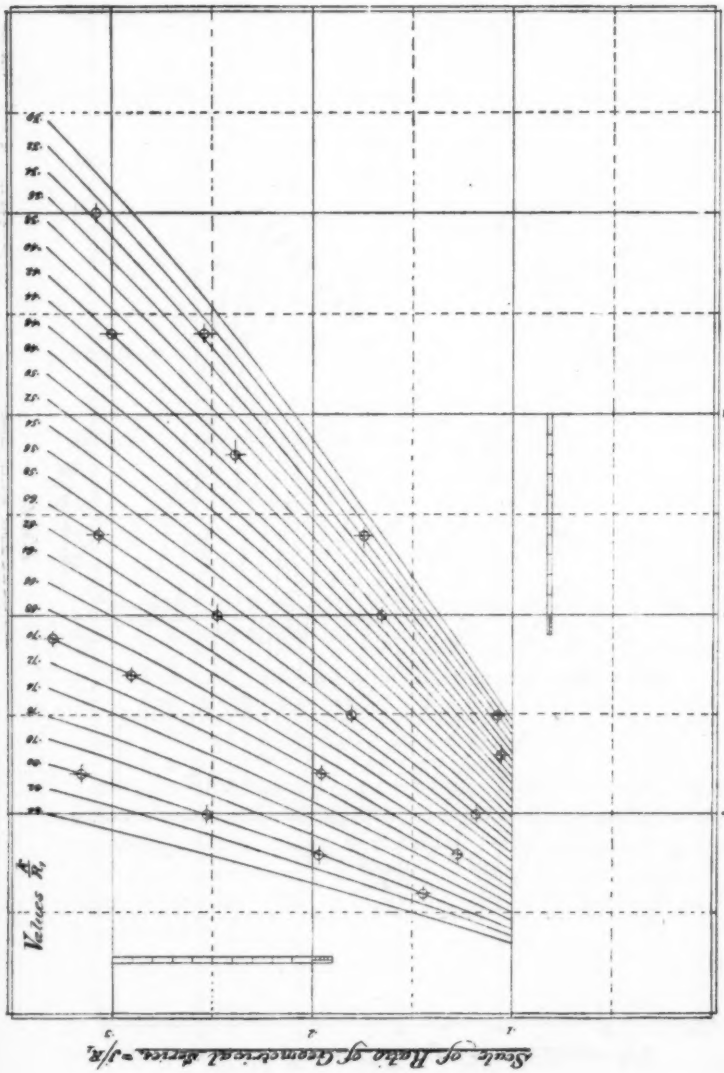

Scale of $\frac{E}{R}$.
Values of $\frac{v}{100}$
Values of $\frac{p}{R}$
Scale of the Ratio of Transmitting Stress $= \frac{p}{R}$

Fig 24.


Values $\frac{v}{R_1}$
Scale of $\frac{v}{R_1}$
Key 2.5.
Scale of Ratio of Geometrical Series $= r/R_1$

where, for the full gear position of OA, b is small compared both with k and a, and ε is a small angle.

The resultant horizontal movement of c_1 could be evolved from equation (97) in the same way as equations (12) to (15) are obtained from equation (96). (See Appendix C.) But the calculation would be very complex and a sufficient approximation for our present purpose can be obtained from Figs. 23 and 24.

If the total vertical movement expressed by equation (97) were small compared with k we could treat the resultant motion of the valve as a superposition of the movements due to $a \cos \theta$ and $b \sin 2(\theta + \varepsilon)$ separately. This would, no doubt, in any practical case give a result much under the truth—possibly several times too small. But it may be used to find if the result on the higher period movements is large or small, and if it increases or decreases as we pass to higher and higher periods.

It will be sufficient to take the case of the *Newark* as there will be no practical cases in which the result is much larger.

Example. It will be shown below—see Table VII—that in the *Newark* we have,—

Natural period of the gear is about $\frac{1}{13}$ second, giving a vibration which is of the 6th period when the engine is in full gear ahead and running full speed—about 127 revolutions per minute. Ratio, for the term $a \cos \theta$, of equivalent geometrical series, Fig. 25 = .305.

Also $R_1 = 21.78$ inclus, $k/R_1 = .765$, and

6th period term of valve motion for rigid gear $= 10^{-3} \times 1.505 \cos 6\theta$, the amplitude being in feet.

Plotting FF_1 we find $b = .14$ inch. Let us take $b = .25$ inch, which will much more than include the effect of the angular motion omitted under (a)—also approximately of 2d period—and the change from k, equation (96) to $k + b$, equation (97).

$$\frac{b}{R_1} = \frac{.25}{21.78} = .01.$$

We can readily see from Figs. 24 and 25—extending the curves in the former to the ordinate $a/R_1 = 0$ and in the latter to the origin, to which all the ratio curves converge—that the

series due to $b \sin 2(\theta + \varepsilon)$, determined by $k/R_1 = .765$ and $b/R_1 = .01$, is

1st term = about $.95R_1 = \frac{.95 \times 21.78}{12}$ feet.

Ratio = about .02.

Therefore, 4th term = about

$$\frac{.95 \times 21.78 \times .02^3}{12} \sin 6(\theta + \varepsilon) = 10^{-3} \times .014 \sin 6(\theta + \varepsilon).$$

The worst case will be when $\sin 6(\theta + \varepsilon) = \cos 6\theta$.

Then the altered amplitude of the 6th period is

$$10^{-3}(1.505 + .014) = 10^{-3} \times 1.519,$$

or a change of less than one per cent.

The maximum effect on the phase of the motion would be when $\sin 6(\theta + \varepsilon) = \sin 6\theta$. Then the tangent of the angular change of the radius Oa, Fig. 14, is

$$\frac{10^{-3} \times .014}{10^{-3} \times 1.505} = .0093.$$

That is, an angular change of $0° - 32'$.

Though these effects were ten times as large they would not, for our purpose, be very important.

Since the ratio, for the term $a \cos \theta$, of the equivalent geometrical series is .305, the fall of amplitude from one even period to the next is $.305^2 = .093$. As this is much larger than the above ratio, .02, the effect of $b \sin 2(\theta + \varepsilon)$ will diminish as we pass to higher and higher periods.

(c) *Rotation about C.*

As the beam F_1CC_1 rotates about C, the speed of the upper and lower ends will be in the proportion

$$\frac{CC_1}{CF_1} = \frac{AA_1}{AS} = \frac{A_1S - AS}{AS} = m - 1.$$

If the centers F_1, C, C_1, are in line—which has been tacitly assumed in the preceding deductions—the movements of the upper and lower ends will, while rotating about C, be at all times op-

positely directed. Hence the total horizontal movements C_1D and EF will be in proportion to their velocities. That is,

$$C_1D = (m-1)\,EF = -(m-1)\,a \sin\theta. \quad . \quad . \quad . \quad (98)$$

Thus, this part of the motion is exactly simple harmonic of the 1st period.

59. REMARKS ON INACCURACIES IN ANALYSIS.

Let us note what is not exact in the above analysis:—

(1) The effect of A_1V and any other intermediate connections to the valve has been disregarded.

If A_1V is comparatively long and if V and O_1 are both on the same side of the tangent, at A_1 to the curve $A_1B_1C_1$, its effect will be to diminish somewhat the amplitudes of higher period. Its best length will depend on ϕ_1, R_1, and the maximum value of vertical movement C_2D, being greater as these quantities are greater. This best length it is of no practical importance to determine. The effect is no doubt of somewhat the same order of magnitude for each period as that due to FF_1, rejected above; for, a moment's reflection will show that the movement it adds to V is principally of second period. If V and O_1 are on opposite sides of the tangent at A_1 the effect will be to increase the amplitudes of higher period, and its effect on those amplitudes will be much larger than for the arrangement in Fig. 26.

In the U. S. S. *Katahdin* a bell-crank lever VGH has been introduced. Its effect will be noted more conveniently along with a few other remarks to be made on the working of this engine in Chapter XII.

(2) The circular arc of radius R_1 has been determined to have contact of the second order—that is, to have the same tangent and radius of curvature—at A_1 with the actual curve $A_1B_1C_1$ described by the upper center when the lower center moves with F. It may readily be shown that, except at the point A, the circular arc lies inside the curve; hence the effect of substituting the circular arc is to raise the estimate of the amplitudes. The epochs will not be altered. However, even when drawn to a large scale the divergence of the circular arc and curve is very slight till practical limits have been exceeded.

(3) Lastly, the omission of the part FF_1 of the movement lowers the estimate of amplitudes in the way fully shown.

The equivalent simple gear deriving motion from F, together with the simple harmonic motion of 1st period, $-(m-1)\,a\sin\theta$, may then be substituted with almost perfect accuracy for the actual gear especially where we are considering the higher period movements, for which also the geometrical series given by Figs. 24 and 25 are practically accurate.

Of course, in considering steam distribution and the inertia for the rigid gear, the exact series must be taken as the lower period terms, for which the geometrical series is sensibly in error, are then the important ones; also the inertia of the term $b\sin 2(\theta+\varepsilon)$ is important. But the true rigid gear curve can be so easily obtained graphically, and from it the inertia for the rigid gear measured, that we need develop no other method.

We have then in the Marshall gear two simple harmonic motions of 1st period affecting B_1, $R_1Q_1\cos\theta$ (see equation (92)), and $-(m-1)\,a\sin\theta$; the variation of ψ_1, as the gear is linked

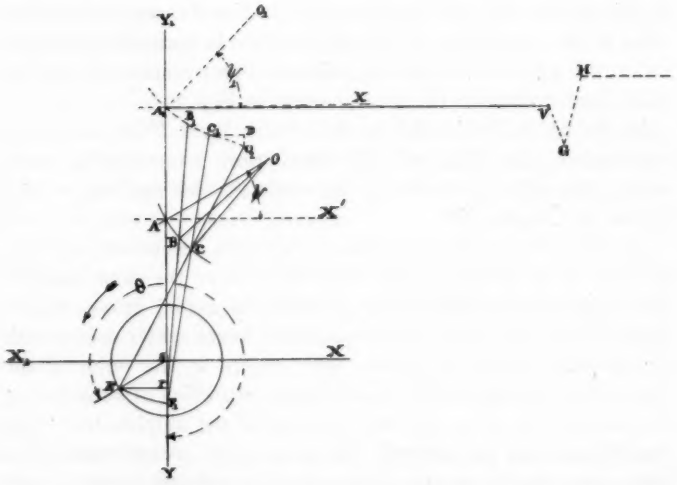


Fig 26.

out, changing their relative importance and making the maximum displacement of simple harmonic motion of the 1st period occur at different crank angles as the cut off changes, in the manner well known.

60. EPOCH OF HIGH PERIOD RIGID GEAR VIBRATIONS.

Combined with these there are movements of all higher periods, the amplitudes rapidly diminishing as we pass to higher and higher periods. These amplitudes also depend on ψ_1 and are extremely sensitive to any change in this angle, as shown by Table III, the response to these changes rapidly increasing as the period becomes higher. These higher periods, except the 2d which is very sensibly affected by the motion FF_1, all come to a maximum to the right, Fig. 26, when F is exceedingly near its lowest point ($\theta = 0$). This follows from equation (12); for if $\theta = 0$ this equation becomes

$$\frac{r}{R} = .7047 - .3546 - .1029 - 10^{-2} \times 2.81 - \&c.,$$

horizontal distances being positive to the left in Fig. 4. This I shall call the *high maximum of* B_1. This is not a point of maximum inertia stress of the valve as the 1st period inertia stress is far from a maximum at this time.

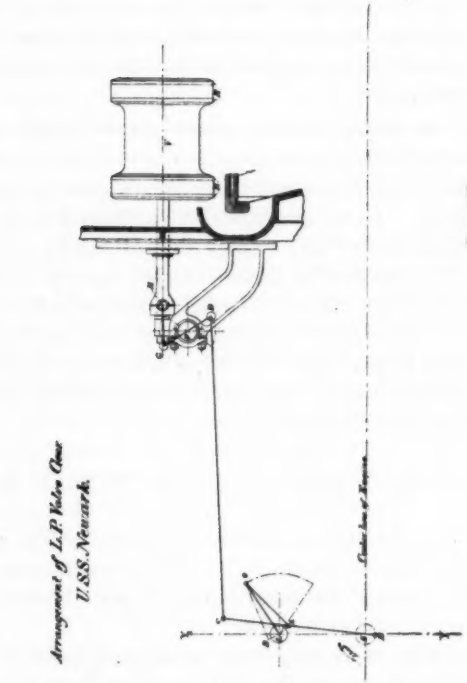
When F, Fig. 26, is very near its highest position the odd period vibrations, except the first, come to their maximum to the left; those of even period, except the second, come to their maximum to the right. For equation (12) becomes, when $\theta = \pi$

$$\frac{r}{R} = .7047 + .3546 - .1029 + 10^{-2} \times 2.81 - \&c.$$

In other words the even periods have, in passing from $\theta = 0$ to $\theta = \pi$, completed a whole number of cycles; the odd periods have completed a whole number plus one-half cycle. This I shall call the *low maximum of* B_1.

If the gear were reversed, bringing O below A, the higher period vibrations would, obviously, all come to their maximum to the right when F was near its highest position.

Thus the *high maximum of* B_1 may be generally described as



Arrangement of L.P Valve Gear U.S.S. Newark.

FIG. 27.

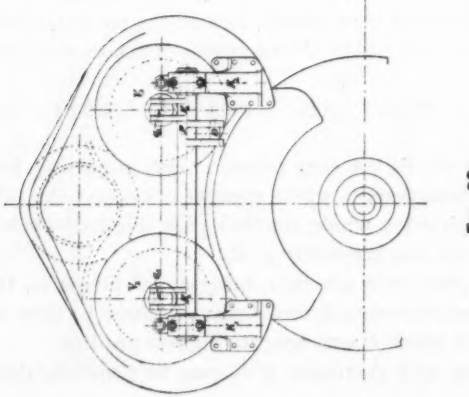


FIG. 28.

that maximum displacement of B_1 from YY_1 due to the movement of F, for which B_1 has its greatest x displacement from A_1.

At the *low maximum of* B_1 it has its least maximum x displacement from A_1.

CHAPTER XI.

APPLICATION OF PRECEDING THEORIES TO U. S. S. *NEWARK'S* L. P. VALVE GEAR.

61. ESTIMATION OF a_6.

Let us estimate the amplitude a_6 of the 6th period rigid gear vibration of the *Newark's* L.P. gear.

Example.

a	$= 3$ inches.
R	$= 18$ inches.
AS, Fig. 26	$= 25\frac{11}{16}$ inches.
A_1S, Fig. 26	$= 45\frac{5}{16}$ inches.
m	$= A_1S/AS = 1.764$.

Table VII shows the calculation from this data for three cut offs.

TABLE VII.—CALCULATION OF 6TH PERIOD RIGID GEAR AMPLITUDE OF U. S. S. *NEWARK'S* L. P. VALVE GEAR.

Cut-off	.76	.725	.65
ψ_1, from working drawings	33°—55′	31°—51′	27°—36′
ψ_1, from equation (94)	49°—52′	47°—37′	42°—41′
R_1, from equation (95)	21.78 inches.	20.42 inches.	17.88 inches.
$b/R_1 = \sin \psi_1$, see Table II	.765	.739	.678
a/R_1	.138	.147	.168
1st term of geometrical series, Fig. 24	1.030 R_1 inches.	.989 R_1 inches.	.924 R_1 inches.
Ratio of geometrical series, Fig. 25	.305	.289	.262
a_6 = 7th term of substituted geometrical series	$\frac{1.030 \times 21.78 \times .305^6}{12}$ $= 10^{-3} \times 1.505$ ft.	$\frac{.989 \times 20.42 \times .289^6}{12}$ $= 10^{-3} \times .981$ ft.	$\frac{.924 \times 17.88 \times .262^6}{12}$ $= 10^{-3} \times .445$ ft.

62. CALCULATION OF THE ELASTICITY OF THE GEAR.

Figs. 27 and 28 show in outline the arrangement of the *Newark's* L.P. gear. There are two valves V_1 and V_2, driven through the shaft F, from one eccentric. The procedure in obtaining the natural period of any gear is first to find the yield of the valve for a given pull on it in its line of motion. To do this we may

take each part of the gear separately, considering for the moment all other parts rigid, but the joints perfectly free to act. Assuming, say, one pound pull on the valve we can from a diagram of the gear readily find the forces acting on the selected part for any configuration of the gear; then the yield of the point of application of the force; lastly, the yield of the valve due to the yield of this part. For instance, in the *Newark's* gear suppose we wish to find the effect of one pound pull on valve V_1 displacing it to the right, arising from the transverse elasticity of shaft F. We may suppose for the moment all the rest of the gear to be rigid and shaft F to have no torsional elasticity. One pound pull on V_1 gives practically one pound tension on link G_1H_1. This, acting through lever G_1F_1, would rotate the shaft were it not resisted by a force induced at D, which is held by the supposed rigid parts of the gear. The intensity of the force at D can be found by taking moments about the center of shaft F. Evidently, the forces at G_1 and D both tend to bend the shaft back toward the cylinder. As we are not considering the torsional elasticity of shaft F we may suppose the forces at G_1 and D applied at F_1 and F, Fig. 28, inducing readily calculated reactions at brackets K_1 and K_2. Applying well-known formulas for the transverse elasticity of beams, and calculating the effects of the forces at F_1 and F, either separately or together, we may find the yield at F_1, F_2, and F, the brackets K_1 and K_2 being rigid. This will give only a part of the yield of the valves; for the point D being fixed and the lever DF rigid, the movement of the point F, Fig. 28, will cause a rigid body rotation of the shaft, which may be treated as a rotation about the practically fixed point D, and the consequent further yield of the valves estimated. The operation need not be described further as it is not difficult to carry out, though requiring reasonable care.

The results for the *Newark* are given in Table VIII. Two configurations of the gear were taken.

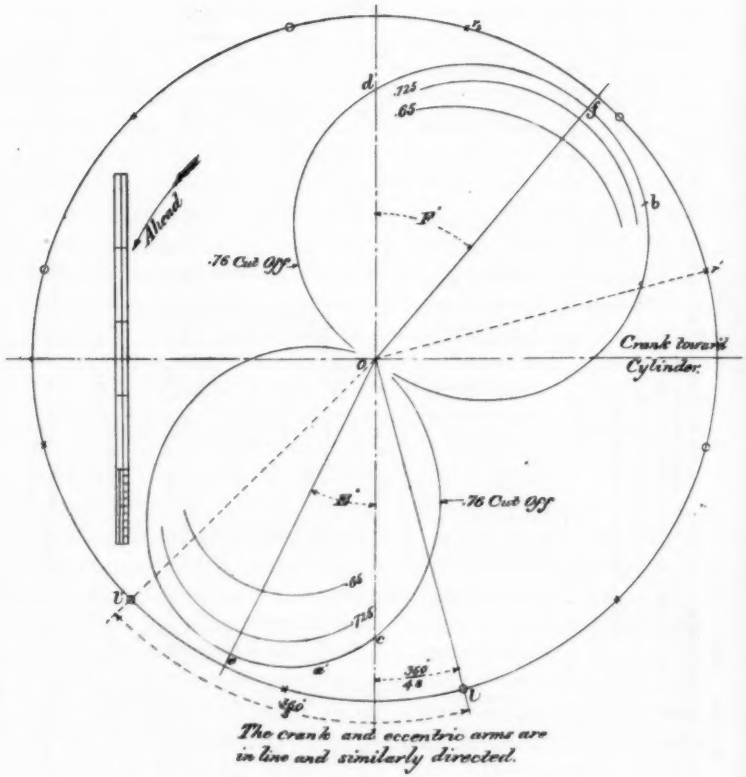
1st. When it was most elastic, which is obviously when the point B, Fig. 26, in the eccentric beam is near its lowest position, as then the stress on the reversing shaft L is greatest and the

TABLE VIII.—ELASTICITY OF L. P. VALVE GEAR, U. S. S. *NEWARK*.

The figures in the table give the yield of the valves in millionths of an inch per pound pull on one valve.
Young's Modulus of Elasticity $= 30 \times 10^6$ pounds per square inch.
Modulus of Rigidity $= 12 \times 10^6$ pounds per square inch.

Part of gear.	Gear in most elastic configuration.								Gear in least elastic configuration.							
	One pound pull on valve V_1.				One pound pull on valve V_2.				One pound pull on valve V_1.				One pound pull on valve V_2.			
	Tension or compression.	Cross-bending.	Torsion.	Consequent yield of valve V_2.	Tension or compression.	Cross-bending.	Torsion.	Consequent yield of valve V_1.	Tension or compression.	Cross-bending.	Torsion.	Consequent yield of valve V_2.	Tension or compression	Cross bending.	Torsion.	Consequent yield of valve V_1.
Valve spindle	.229	...	...	...	.229	...	...	...	.229	...	...	...	.229	...	...	...
Link GH	.060	...	...	...	.060	...	...	...	.060	...	...	...	.060	...	...	...
Lever FG	...	.064	...	...	...	.064	...	...	...	.064	...	...	...	.064	...	...
Shaft F	...	...	.381	...	...	...	3.123	...	...	...	.381	...	...	...	3.123	...
Shaft F	...	10.323	...	9.521	...	8.961	...	9.064	...	10.323	...	9.521	...	8.961	...	9.064
Bracket K_1	.063	...	...	.008	.004	...	...	.034	.063	...	...	.008	.004	...	...	.034
Bracket K_2	.002	...	...	.012	.041	...	...	.005	.002	...	...	.012	.041	...	...	.005
Lever FD	...	.038	...	.038	...	.038	...	.038	...	.038	...	.038	...	.038	...	.038
Link CD	.571	...	...	.571	.571	...	...	.571	.571	...	...	.571	.571	...	...	.571
Eccentric beam CE	...	1.235	...	1.235	...	1.235	...	1.235	...	1.235	...	1.235	...	1.235	...	1.235
Link AO	.375	...	...	.375	.375	...	...	.375	.205	...	...	.205	.205	...	...	.205
Lever OL	...	.981	...	.981	...	.981	...	.981	...	.535	...	.535	...	.535	...	.535
Lever OL	...	...	.223	223	...	...	.223	.223	...	...	...	...	...	...	...	...
Shaft L	...	...	4.650	4.650	...	...	4.650	4.650	...	...	...	...	...	...	...	...
Shaft L	...	8.416	...	8.416	...	8.416	...	8.416	...	4.534	...	4.584	...	4.584	...	4.584
Connection to Reversing Engine	.991	.581	7.153	8.725	.991	.581	7.153	8.725	...	...	..	..	...	...	...	...
Total tension or compression	2.291	...	...	...	2 271	...	...	...	1.130	...	...	...	1.110	...	...	...
Total cross-bending	21.638	21.638	...	...	20.276	20.276	...	...	16 779	16.779	...	...	15.417	15.417	...	...
Total torsion	12.407	...	12.407	...	15.149	...	15.149	...	.381	...	.381	...	3.123	...	3.123	..
Total yield of valves	36.336	...	...	34.755	37.696	...	...	34.317	18.290	...	...	16.709	19.650	...	...	16.271

Fig. 29.



L.P. Valve Diagram.

U.S.S. Newark.

yield of the valves for a given rotation of the reversing lever *OL* is also the greatest.

2d. When the gear is stiffest. This is when *OB*, Fig. 26, and *OL*, Fig. 27, are in line so that no torsion comes on the reversing shaft *L* and consequently no stress on the gear connecting it to the reversing engine.

Table VIII shows that the stiffness of the gear in the latter case is about twice that in the former. This varying stiffness of the gear would be exceedingly difficult to take exact account of, but we may calculate the natural period for the stiffest and for the most elastic configurations. There is some period between these which, if made to synchronize with one of the rigid gear vibrations, will undoubtedly give rise to results like those deduced on the supposition of constant elasticity. It seems most probable that the elasticity, which would give this intermediate period is nearer to that for the most elastic than for the least elastic configuration; since, as the engine revolves, the gear remains much longer near its maximum than near its minimum elasticity. As the natural period is in inverse proportion to the square root of f—the rigidity of the gear—the mean of the two periods calculated for greatest and least elasticity will correspond to a value of f lower than its mean value, and we will certainly be within the limits of error of the calculation by taking this as the natural period.

63. REMARKS ON *TABLE VIII.* ADVANTAGE OF STEPHENSON'S GEAR. DR. KIRK'S EXPERIENCE.

I have given the calculated results of the elasticity of the gear in detail to show how much the total yield is due to crossbending and torsion and how little to tension and compression.

This brings out the true reason of the great success of the Stephenson link motion for high speed engines. As usually fitted, it has a very direct connection between the working eccentric and the valve, composed almost entirely of parts under tension and compression; therefore, the natural period is very high.

In every gear where there is danger of serious inertia stresses, the parts under crossbending and torsion should be determined to give ample rigidity. The ordinary calculations of strength

are not sufficient. We will see what large inertia effects were due to the elasticity of the *Newark's* gear; and we will be better able to judge from the sequel what is a safe degree of elasticity and what total stress should be provided for.

I recollect Dr. Kirk remarking to me many years since that he had given up working valves through shafts under torsion. This must have been the result of his experience in Messrs. John Elder and Company's in the '70's, as I am familiar with all his subsequent work. His remark was made in connection with the question of the stresses induced, and shows how long ago the effects with which we are dealing were noticed in connection with valve gears.

64. CALCULATION OF NATURAL PERIOD AND *S*.

Table VIII shows,—

1st. That the rigidities of V_1 and V_2 are always nearly the same.

2d. That a pull on V_1 produces nearly the same displacement of V_2 as of V_1, and vice versa.

If the rigidities of the two valves had been exactly equal and if, also, the consequent yield of the unpulled valve had been the same as that of the one pulled we could have treated the two valves as one. This simple case is what is given here, as it is that always to be used when there is a single valve. The more exact treatment of the *Newark's* gear is given in Appendix E. It is much more complex, but as there may be cases in which the rigidities and consequent yields of the two valves are very different it seems necessary to give the treatment for this case. It is there shown that instead of one natural period there are a pair, which might be called the principal natural periods; for, it requires little insight to see that, could the exact problem be solved, and the influence of each separate part of the gear taken into account, the number of natural periods would be greatly multiplied. In fact one cannot help calling to mind the analogous phenomena of many natural periods, exhibited by the spectrum, some of great relative energy giving markedly bright lines, but accompanied by a host of others less bright; the phenome-

non being, no doubt, due to the complex structure of the molecule, giving many different elasticities and times of vibration.

The treatment of the gear driving two valves will also show how even more complex cases should be calculated. These will not frequently occur in connection with valve gears but may in other applications of the same theory to high speed machinery.

Example.—Approximate calculation of the natural period of *Newark's* L.P. valves and the number of the synchronizing period at 127 revolutions per minute.

	Gear in most elastic configuration.	Gear in least elastic configuration.
M = mass of two valves, stems and part of gear.....	2360 lbs.*	2360 lbs.*
f_1r = pull on valve.....	1 lb.	1 lb.
$_1r$ = mean yield of valve in feet. (See Table VIII)..	$\frac{1}{4}\times\frac{10^{-6}}{12}\times\begin{cases}36.336\\34.755\\37.696\\34.317\end{cases}$	$\frac{1}{4}\times\frac{10^{-6}}{12}\times\begin{cases}18.290\\16.709\\19.650\\16.271\end{cases}$
	$=\frac{10^{-6}}{4\times12}\times 143.104$	$=\frac{10^{-6}}{4\times12}\times 70.920$
$f=1/_1r$.....	$=10^{-6}\times 2.981$ feet.	$=10^{-6}\times 1.477$ feet.
	$10^6\times .3355$ lbs.	$10^6\times .6770$ lbs.
$\omega_s=\sqrt{\frac{fg}{M}}=\sqrt{\frac{32.2f}{2360}}$. See equation (17)	67.65 radians per second.	96.11 radians per second.
T = natural period $=\frac{2\pi}{\omega_s}$. See equation (16).....	.09287 sec.	.06537 sec.
Number of vibrations per minute $=60/T$.....	646.0	917.8
Number of natural period of vibration when the engine is running 127 revolutions per minute.	$\frac{646.0}{127}=5.087$	$\frac{917.8}{127}=7.227$
i = number of synchronizing period at 127 revolutions..	= Say $(5.087+7.227)\div 2=6.157$	

* This is a little larger than the value of M taken in *Examples I and II*, § 10. If the whole gear is vibrating with the natural period, the energy of each part is proportional to the product of its mass into the square of its amplitude. The sum of the energy of all these separate parts makes up the total energy which must be equal to the energy expended in displacing the valve when starting the vibration. Hence we see that we arrive at the same natural period if we add to the valve mass that of each part reduced in the proportion of the square of the ratio of its amplitude to the amplitude of vibration of the valve, and regard the gear itself as massless. These amplitudes of vibration evidently bear nearly the same ratio to one another as the yields of the same parts per pound pull on the valve; for the principal inertia arises from the valve, that being much the heaviest part. The change made by this requirement is usually slight and need only be made approximately, if at all.

Practically we may say that the synchronizing period is the 6th.*

If R_s is the number of revolutions of the engine per minute, we have directly,

$$s = \frac{\omega_s}{\omega} = \frac{1}{\omega}\sqrt{\frac{fg}{M}} = \frac{30}{R_s\pi}\sqrt{\frac{g}{M\partial r}}, \quad . \quad . \quad . \quad (99)$$

the mean value deduced from the two values of ∂r being taken, as above.

65. DATA FROM OFFICIAL REPORTS.

TABLE IX.—DATA FROM OFFICIAL REPORTS OF TRIALS U. S. S. *NEWARK*.

1st trial, 10th Dec., 1890. Starboard engine broke down. No data.

2d trial, 12 Dec., 1890. Starboard engine broke down after about 2½ hours at full speed. The Official Report gives only the mean of ten quarter hour periods.

	Starboard engine.	Port engine.
Revolutions per minute	127.03	126.41
Cut-off in decimal of stroke	.75 H.P., .76 I.P., .725 L.P.	.72 H.P., .72 I.P., .65 L.P.
Absolute pressure in 2d receiver	31 lbs.	33.5 lbs.
Vacuum	25.4 inches.	25.48 inches.
Barometer	30.02 inches.	

3d trial, 22d Dec., 1890.

		Starboard engine.	Port engine.
Revolutions per minute	highest.	129.5	128.7
	lowest.	124.0	124.0
Cut-off in decimal of stroke		.76 H.P., .76 I.P., .62 L.P.	.72 H.P., .72 I.P., .65 L.P.
Absolute pressure in 2d receiver	highest.	32.5 lbs.	31.7 lbs.
	lowest.	30.5 lbs.	30.7 lbs.
Vacuum	highest.	26.6 inches.	26.0 inches.
	lowest.	25.5 inches.	25.5 inches.
Barometer	highest.	30.37 inches.	
	lowest.	30.34 inches.	

NOTE.—The parts which broke on the 1st and 2d trials were the two 1⅜ inch mild steel studs at the end B of the radius link OB, Fig. 27. Before the 3d trial a new radius link was made having 1¾-inch studs of Norway iron.

* For remarks on the effect of the variability of f see Appendix D.

66. ESTIMATE OF PROBABLE VALUE OF F.

Each L.P. valve has four split bronze rings R, Fig. 27, working on the valve seat. These are dimensioned in the working drawings to be turned 30 inches outside diameter—the exact inside diameter of the valve chamber. Behind each is another split ring dimensioned $\frac{1}{32}''$ larger than its diameter when in place. Thus the friction from the spring of these rings, even before there had been any wear, would be very small.

In any position of the valve the full receiver pressure acts on the plain vertical face of two of these four outer rings, the total force acting parallel to the axis of the valves being much greater than the friction. This tends to open a passage for the receiver steam to the small space behind the rings and shut off the escape of this steam to the exhaust side. The rings have to be left with freedom to move, and however well the contiguous surfaces of the valve and rings were scraped and fitted no doubt there would be some leakage. This might raise the pressure behind the rings to the receiver pressure, but as it is unlikely that all leakage to the vacuum side would be cut off entirely we cannot count on quite so high pressure as this even at first.

If there were no vacuum and the receiver pressure got freely at both inboard and outboard sides of any ring, this pressure at the back would evidently not produce any normal pressure between the ring and valve chamber. But if now we suppose the vacuum formed on one side, a pressure gradient will be established in the vapor or moisture between the ring and chamber. The flow near the vacuum side will intensify the pressure on any moisture left nearer the pressure side. The ultimate result will be the almost instant production of very close contact between the metal of the ring and chamber.

Thus the highest estimate of normal pressure which can be made is that the difference between the receiver and vacuum pressures is acting behind two rings in each valve. But this is probably in all cases somewhat too high; and if the contiguous surfaces of the valves, rings and chamber are not kept in good condition this assumption may be much too high.

To this must be added the weight of the valves, which in the *Newark* slide horizontally.

By equation (26) the coefficient of friction lies between .3 and .14 if the surfaces are dry or wet with water. If slightly greasy —as they would almost certainly be, at least on a trial trip—the coefficient of friction might be considerably under .14:

Thus for two valves we have,—

Four rings, 30 inches diameter by $\frac{1}{8}$ inch wide. Area of face	=	329.9 sq. inches.
Difference of vacuum and receiver pressure from *Table IX*	=	say, 28.5 lbs.
Normal pressure due to steam not greater than 329.9×28.5	=	9402 "
Normal pressure due to weight of valves. *Examples I and II* §10.	=	2232 "
Total normal pressure	=	11634 "
F with coefficient of friction $= .3$	=	3490 "
F with coefficient of friction $= .14$	=	1629 "

It is more probable that the surfaces were slightly damp than that they were dry as they were very near parts of the cylinder exposed to lower steam pressure than that in the receiver. Hence, according to Bochet's experiments the coefficient of friction was probably near the lower of the two limits given above.

As the pressure behind the rings would most likely be lower than the receiver pressure and also the valve face was almost sure to be greasy on trial it is possible that F may not have been above half the lower of the above values. Hence,

$$F = 800 \text{ lbs.}$$

is also regarded as probable in the calculations which follow.

67. DETERMINATION OF ENERGY TRANSFER CURVES AND NORMAL EPOCH.

Fig. 29 shows the *Newark's* full ahead valve curves and parts of those for .725 and .65 cut off. The points a' and b are those of maximum stress calculated in *Examples I and II*, §10.

As the crank and eccentric arms are in line and similarly di-

rected, Fig. 27 shows that Oe of Fig. 29 gives the valve displacement to the left, and Of to the right. Also Oc is the position of the crank at high maximum of B_1, Fig. 26, and Od at low maximum.

The rigid gear synchronizing vibration of the valve comes to its left maximum at Oc, but the actual vibration of this period when it has its normal epoch is quarter a period later, which corresponds to a crank angle of 15°, since the 6th period synchronizes.

Therefore mark l with a small circle making $cOl = 15°$. We can now divide off the surrounding circle with small circles and crosses as was done in Fig. 20. Ol' is the crank position for the last left maximum of the actual synchronizing vibration of normal epoch before Oe; and Or' the last right maximum before Of. As shown in § 49, the difference of the angles $r'Of$ and $l'Oe$ is 16° and the zero ordinate of the n_e curve, Fig. 21, must be superposed on the $16 \times 6 = 96°$ ordinate of the n_f curve, and the mean curve between these two drawn. This gives the curve *aaaa*, Fig. 30.

Angle $lOc = 15°$.
" $lOl' = 60°$.
Therefore " $l'Oc = 45°$.
" $eOc = 26°$.
Therefore " $eOl' = 19°$.

Thus we get,

Normal epoch of actual synchronizing vibration, curve *aaaa*, Fig. 30, $\Big\} = 19 \times 6 = 114°$

This ordinate is marked AA. It is the ordinate at which energy transfer due to friction must be measured when the actual synchronizing vibration has its normal epoch, and the 114° is measured from the $O°$ ordinate of the n_e curve.

68. DETERMINATION OF F FOR $n_e = 2.0$ — FULL AHEAD GEAR.

As stated in § 49, curve *aaaa*, Fig. 30, is deduced from $n_e = 2.00$ and $n_f = 3.3$. Let us determine the value of F in order that I_1 may remain at the corresponding value.

The normal epoch cannot be retained, as shown in §50. As the singular point a in Fig. 30 gives a minimum rate of absorption of energy, let us first assume—what is very nearly true—that it is the actual epoch required. (Fig. 30_1 shows that the actual epoch is reached for a rather smaller value of the angle a.)

We have, then, for the change from normal epoch,

$$a = 25°.$$

From Fig. 30, $\tau = 5.84$ at this point.

From equation (90) we have,

$$F = \frac{\pi f s a_s}{\tau} \cos a.$$

The fairest value of f to take in this equation is probably that corresponding to $s = 6.157$, given in § 64. It may be calculated from the value of ω_s given there. Thus,

$$f = \left(\frac{67.65 + 96.11}{2}\right)^2 \times \frac{2360}{32.2} = 491{,}374 \text{ lbs.}$$

$$s = 6 \text{ and } fs = 2{,}948{,}244.$$

From *Table VII*, $a_s = 10^{-3} \times 1.505$ feet.

Using these values in equation (90) we get,—

$$F = \frac{\pi \times 2.948{,}244 \times 10^{-3} \times 1.505 \times \cos 25°}{5.84} = 2163 \text{ lbs.}$$

This value of F is much above the probable value — See § 66.

69. STRESS IN STUDS WHICH BROKE.

For a lower value of F, unless there was a breakdown, a_s would increase till I_s was far above $2I_c$. But when $n_s = 2$, the stress in the studs which broke is not far from the greatest they could withstand. The maximum tension will come on the link OB, Fig. 27, and on the 1⅛ studs which broke, when the crank is between Od and Of, Fig. 29. Then, it can readily be deduced from Fig. 27 that, one pound tension on the valve spindle will put a stress of from .9 to 1.0 pound per square inch on the area under the thread of each stud, and Fig. 29 shows that inertia from the actual synchronizing vibration and friction will then act along with the inertia due to the non-synchronizing terms.

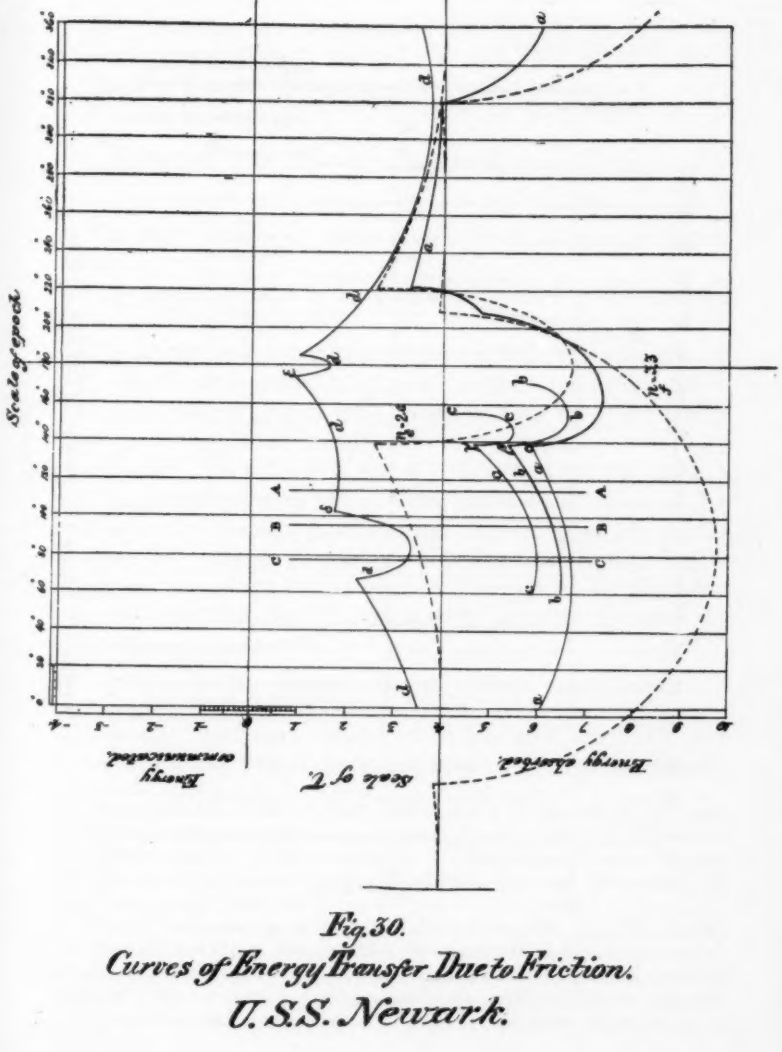


Fig. 30.

Curves of Energy Transfer Due to Friction.

U.S.S. Newark.

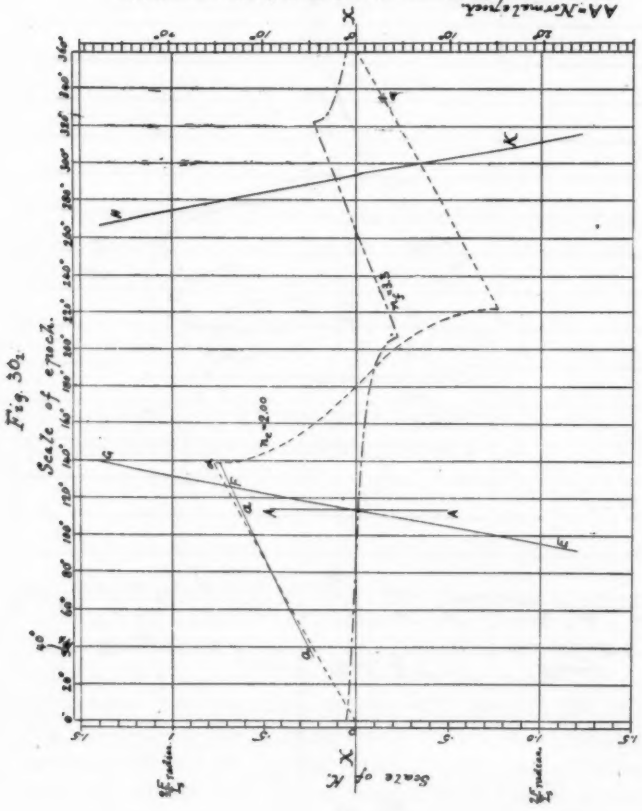

AA = Normal epoch.
Curve aa gives rate of increase of epoch due to friction, per reaction.
Curve EfGhK gives rate of variation of normal epoch due to energy loss.
Like gears, per reaction.
Scale of epoch.
Scale of R_2
$R_2 = 2.00$
$R_2 = 1.33$

Fig. 30₂

Curves of Change of Epoch, 70 Cut Offs $R_2 = 2.00$.

U. S. S. Newark.

Hence the stress under the threads of the 1⅛ inch mild steel studs is, very nearly,—

$I_1 = 2 \times$ stress at e, Fig. 29, = $2 \times 6{,}290$, *Table X*, line (m)	=	12,580 lbs. per sq. in.
Stress at f, *Table X*, line (n) . . .	=	3,813 " " " "
F_1 as determined above	=	2,163 " " " "
Total stress	=	18,556 " " " "
	=	8.28 tons " " "

The stress is certainly above $8.28 \times .9 = 7.45$ tons per square inch.

(The direct addition of these separate stresses to find the total stress is not strictly correct.

The normal epoch has been departed from by an angle $a = 25°$. This corresponds to a crank angle $a/s = 25/6 = 4.25°$. That is, the angle $l'Oe$, Fig. 29, increases by 4 25° ; and, consequently, the angle r_2Of decreases by this amount.

Obviously, the maximum stress occurs between the radii Of and the shifted position of Or_2. The position and value of this maximum can readily be determined by plotting on Fig. 29 the values of F, of the varying inertia from the actual synchronizing vibration, and of the inertia from the curve bfd. As the latter changes little between Of and Or_2 it is sufficient to make the direct division as above.)

70. INTENSITY OF STRESS WHICH WOULD PROBABLY DETERMINE FRACTURE.

In testing mild steel permanent set has been noted at as low a stress as about 8 tons per square inch* and immediately after

* See *Experiments on Riveted Joints* by Prof. Alex. B. W. Kennedy in the *Transactions of the Institution of Mechanical Engineers*, London, 1881. His Fig. 1 shows the magnified diagram of sets of a specimen of boiler plate steel—harder than the steel from which bolts are usually made—breaking at 29.40 tons. Permanent set is noted as beginning at 8.21 tons.

To quote Wiedemann's paper *On Torsion* already cited (*Phil. Mag.* 1880, page 2), "as Wertheim had already observed, the former (permanent deformations of a body) occur even after the slightest temporary deformations, so that in a body not previously deformed a limit of elasticity does not properly exist, but bodies which have sufficiently often been deformed within certain limits by determinate forces, on the re-

sensible extension starts a heavy hammering action would be set up, greatly increasing the stress on these studs and producing almost instant breakdown of the gear.

There is, however, a great difficulty in determining the stress which should be considered unsafe in this case. We must remember that the pressure changes from one brass to the other when the total stress has just become zero, and this will mitigate the effect of the hammering when the joint begins to open through extension of the studs. But it is only possible to run such a gear at considerable speed when all bearings are very carefully set up with but small clearance, thus proving the large effect of the exceedingly small slacking of the brasses. Much may depend on the particular case under consideration, and it is always probable that the stress is not coming quite fairly and equally on all studs or other parts in danger of breakage. Careful experiment will always show considerable set in mild steel at 12 tons per square inch, though the stress at the elastic limit is usually put down as

---

peated action of the same or less forces in the same direction as that last applied, constantly again receive the same permanent deformation."

Again (see Todhunter and Pearson's *History of Elasticity*, Vol. II, Art. 710.) "Wiedemann remarks that it depends merely on the sensibility of our apparatus whether or not we are able to measure the set of the smallest loads."

In the case considered in the text a heavy initial stress would be put on the studs when screwing up the nuts. The pressure between the bearing cap and body, thus induced, would have to be relieved before the stress on the studs could be sensibly raised.

It is interesting to note that we have here a case, belonging to a large class, in which the ordinary mode of determining the factor of safety from the breaking strength of the material is entirely devoid of meaning. The very slightest permanent extension so changes the conditions that it absolutely determines fracture.

I think a careful examination of Prof. Kennedy's Fig. 1 will lead to the conclusion that fracture would almost certainly follow a stress of about 12 tons per square inch on the studs, and possibly follow a stress of not much over 8 tons. This point would be most difficult to determine with accuracy. Had the friction of the *Newark's* L.P. valves, under the trial conditions of 2d receiver pressure, and vacuum, been determined experimentally, the second unsuccessful trial would have given means of determining this point, as the engine then ran for several hours before breaking down, showing that the extreme practical limit was not far passed.

Wöhler's experiments determine a far higher limit of fracture when the stress varies between a maximum and zero, as in them permanent extension does not radically alter the whole conditions and stresses. See *Tables XIV* and *XV*.

higher than this. Twelve tons per square inch seems the highest possible value we can put on the safe stress in such cases as the present, and it would be much more prudent, in the absence of direct experiment, to put it at 8 tons per square inch.

Considering the factors of safety ordinarily applied, even 8 tons per square inch may seem far too high a limit. But as an engineer gains experience and is led to examine minutely the actual stresses which may come on parts of machinery and which it is most difficult or sometimes impossible to estimate (such as cooling stresses in castings, hammering and forging, flaws, want of alignment in bearings, and vibrations due to elasticity and inertia, &c., &c.) he becomes convinced that in many cases of successful machinery the elastic limit must be very closely approached. The examination, in the last chapter, of some of Wöhler's experiments will bear this out very strongly.

71. CALCULATION OF STRESS FOR .76, .725 AND .65 CUT OFF.—*TABLE X.*

Table X gives all the steps in the above calculation of stress for .76 cut off and also those for .725 and .65 cut off; the latter not only for $n_e = 2.00$, but also for $n_e = 1.2$.

The way in which each result is obtained has been carefully indicated in the table and foot notes, so that no further explanation is required. This also makes the table an index of much that has gone before and a guide to any future calculations of the same kind.

72. REMARKS ON RESULTS IN *TABLE X.*

In no case are J and J', *Table X*, very small. Consequently the changes, during the revolution, of the elements of the actual synchronizing vibration may be considerable. But the restricted theory of *Chapter VIII* still applies to this case very fairly, and it is possible to predict the safe angle of the reversing with great accuracy as the effects of a slight angular change in it are so marked.

The results of an examination of the table are,—

.76 cut off, and $n_e = 2.0$.

TABLE X.—CALCULATIONS OF THE STRESSES ON TRIAL, AND OTHER

Line.	Quantity.	How obtained.
(a)	Cut-off	Official Report, Table IX
(b)	$n_s = I_s/I_e$	Assumed
(c)	E^{0*}	Measured from Fig. 29
(d)	F^{0*}	
(e)	r_e^*, in feet	
(f)	r_f^*, in feet	
(g)	ρ_e^*, in feet	
(h)	ρ_f^*, in feet	
(i)	$-\frac{d^2r}{d\theta^2}$ at $e = r_e\left(\frac{r_e}{\rho_e} - 1\right)$	Equation (6); lines (e) and (g).
(j)	$-\frac{d^2r}{d\theta^2}$ at $f = r_f\left(\frac{r_f}{\rho_f} - 1\right)$	Equation (6); lines (f) and (h).
(k)	Revolutions of engine per minute	Official Report
(l)	$\frac{M\omega^2}{g}$	§ 10. *Examples I and II*
(m)	I_e	Line (l)×(i). Equation (2)
(n)	I_f	Line (l)×(j). Equation (2)
(o)	$n_f = n_e I_e/I_f$	Equation (91)
(p)	s	§ 64
(q)	fs	§ 68
(r)	Angle $l'Oe$, Fig. 29 $= \frac{3}{4} \times \frac{360}{s} - E^{0*}$	Lines (c) and (p)
(s)	Angle $r'Of$, Fig. 29 $= \frac{5}{4} \times \frac{360}{s} - F^{0*}$	Lines (d) and (p)
(t)	Angle $r'Of$—angle $l'Oe$	Line (s)—line (r)
(u)	Ordinate of n_f curve, Figs. 16_2 and 21, which must be superposed on the O^0 ordinate of the n_e curve=Line (t)×s.	§§ 49 and 67
(v)	Normal epoch=ε_s''=angle $l'Oe \times s$ for Marshall gear.	§ 67
(w)	Normal ordinate, Figs. 30, 30_1, 30_2, 30_3, 30_4	Line (v)
(x)	Energy transfer curves due to friction, Fig. 30	Fig. 21 and lines (b), (o) and (u).
(y)	Curves of change of epoch per revolution of engine, Figs. 30_1, 30_2, 30_3, 30_4.	Fig. 16_2 and lines (b), (o) and (u). § 42 and 51.

PARTICULARS OF THE LOW PRESSURE VALVE GEAR U. S. S. *NEWARK*.

1st trial 10th Dec., 1890.	2d trial 12th Dec., 1890.	3d trial 22d Dec., 1890.	3d trial 22d Dec., 1890.
.76?	.725	.65	.65
2.0	2.0	2.0	1.2
26°	29°	32°	32°
40°	42°	46°	46°
4.425/12	4.11/12	3.5/12	3.5/12
4.425/12	4.23/12	3.92/12	3.92/12
1.85/12	1.72/12	1.43/12	1.43/12
2.4/12	2.22/12	1.94/12	1.94/12
.513	.476	.422	.422
.311	.319	.334	.334
127?	127	127	127
12261	12261	12261	12261
6290 lbs.	5836 lbs.	5174 lbs.	5174 lbs.
3813 lbs.	3911 lbs.	4095 lbs.	4095 lbs.
3.3	2.98	2.52	1.51
6	6	6	6
2,948,244 lbs.	2,948,244 lbs.	2,948,244 lbs.	2,948,244 lbs.
19°	16°	13°	13°
35°	33°	29°	29°
16°	17°	16°	16°
96°	102°	96°	96°
114°	96°	78°	78°
AA	*BB*	*CC*	*CC*
aaaa	*bbbb*	*cccc*	*dddd*
aa and *EFGHK*.	*bb* and *EFGHK*.	*cc* and *EFGHK*.	*dd* and *EFGHK*.

TABLE X.

Line.	Quantity.	How obtained.
(z)	a=departure of actual epoch from normal epoch.	See foot note †† .
(aa)	F=friction corresponding to assumed value of n_s.	
(bb)	a_s, in feet	§ 61. *Table VII* . .
(cc)	τ	Fig. 30, for corresponding value of a.
(dd)	κ	Figs. 30_1, 30_2, 30_3, 30_4, for corresponding value of a.
(ee)	Stress, under threads, in studs which broke. Due to $I_s=n_cI_c$, lbs. per sq. inch	Lines (b) and (m) .
(ff)	Due to I_f, lbs. per sq. inch .	Line (n) . . . .
(gg)	Due to F, lbs. per sq. inch .	Line (aa) . . . .
(hh)	Total stress, lbs per sq. inch .	. . . . .
(ii)	Total stress, tons per sq. inch	. . . . .
(jj)	$J=\frac{F}{I_c}$	Lines (aa) and (m) .
(kk)	$J'=\frac{F}{I_s}$	Lines (aa) and (ee) .
(ll)	Angle eOh, Fig. 17 $=sin^{-1}2F/I_s=$ maximum possible change of epoch of the actual synchronizing vibration, at reversal.	§ 41 and line (kk) .
(mm)	$\frac{2F}{I_s}\times 57.3^0$=degrees of right-hand scale per unit of left-hand scale, Figs. 30_1, 30_2, 30_3, 30_4.	Equation (81); line (kk)
(nn)	$\frac{\pi fsa_s}{I_s}$ = Amplitude of sine curve $EFGHK$, Figs. 30_1, 30_2, 30_3, 30_4, in radians.	Equation (84); lines (q), (bb), (ee).
(oo)	$\frac{a'_s}{a_s}=I_s\div\frac{M\omega^2}{g}-s^2a_s=$ magnification of amplitude of rigid gear synchronizing vibration.	Lines (l), (p), (bb), (ee); § 75.
(pp)	a'_s=Amplitude of actual synchronizing vibration, in feet.	Line (oo)×line (bb) .
(qq)	$\frac{.002fsa'_s}{\tau F}$ Ratio of energy of actual synchronizing vibration absorbed by viscosity to that absorbed by friction.	Equation (108); lines (q), (cc), (pp), (gg).
(rr)	$\frac{I_s}{F}$	Line (kk) . .

CONTINUED.

1st trial 10th Dec., 1890.	2d trial 12th Dec., 1890.	3d trial 22d Dec., 1890.	3d trial 22d Dec., 1890.
25°	10°—46′	9°—22′	11°—50′
2163	1463	687	1240
$10^{-3}\times 1.505$	$10^{-3}\times .981$	$10^{-3}\times 445$	$10^{-3}\times .445$
5.84	6.1	5.92	3.25
††	.58	.488	.34
12580	11672	10348	6209
3813	3911	4095	4095
2163	1463	687	1240
18556	17046	15130	11544
8.28	7.61	6.75	5.15
.3439	.2507	.1328	.2397
.1719	.1253	.0664	.1997
20°—7′	14°—31′	7°—38′	23°—33′
19.70°	14.36°	7.61°	22.89°
1.108	.778	.398	.664
18.94	26.96	52.68	31.61
$10^{-2}\times 2.850$	$10^{-2}\times 2.645$	$10^{-2}\times 2.344$	$10^{-2}\times 1.407$
.0133	.0175	.0340	.0206
5.82	7.98	15 06	5.01

TABLE X—CONTINUED.

FOOT NOTES.

* In these measurements from Fig. 29 the left and right maximum displacements are designated by e and f for all three cut-offs.

E^0 is the angle between Oe and the maximum vector Oe, Fig. 29.

F^0 is the angle between Od and the maximum vector Of, Fig. 29.

r_e and r_f are the lengths of the maximum vectors.

ρ_e and ρ_f are the radii of curvature at the ends of the maximum vectors.

$s = 6$ is retained when the cut-off is shortened, as the stiffest configuration of the gear only gave $s = 7.227$. See *Example*, § 64.

✡ Take the smallest positive angle of the series—

$$\frac{3}{4}\times\frac{360^0}{s}-E^0, \frac{7}{4}\times\frac{360^0}{s}-E^0, \frac{11}{4}\times\frac{360^0}{s}-E^0, \text{ \&c.}$$

† When s is even take the smallest positive angle of the series—

$$\frac{1}{4}\times\frac{360^0}{s}-F^0, \frac{5}{4}\times\frac{360^0}{s}-F^0, \frac{9}{4}\times\frac{360^0}{s}-F^0, \text{ \&c.}$$

When s is odd take the smallest positive angle of the series—

$$\frac{3}{4}\times\frac{360^0}{s}-F^0, \frac{7}{4}\times\frac{360^0}{s}-F^0, \frac{11}{4}\times\frac{360^0}{s}-F^0, \text{ \&c.}$$

†† .76 cut-off. α assumed to give the epoch of the actual synchronizing vibration the same as that of the singular point α, Fig. 30, which gives a minimum value of τ. This value of α is actually rather large. F is then calculated from equation (90),

$$F = \frac{\pi f s a_s}{\tau} \cos \alpha.$$

Thus, the 1st condition of § 38 is not quite exactly fulfilled for this cut-off and κ is omitted from the table.

.725 and .65 cut-off. α and F are determined by trial and error, so as to fulfil the 1st and 2nd conditions of § 38. That is,

$$F = \frac{\pi f s a_s}{\tau} \cos \alpha, \text{ equation (90),}$$

and

$$F = \frac{\pi f s a_s}{2\kappa} \sin \alpha, \text{ equation } (91_2)$$

simultaneously; and therefore also

$$\tan \alpha = \frac{2\kappa}{\tau}, \text{ equation } (91_3).$$

The calculated values of α are given to minutes, merely to fulfil the above approximate equations. But, actually, a change of epoch of a considerable number of degrees may take place at reversal, as shown by line (11).

The angle α may be found very simply by trial and error—

1st. Assume a value of α.

2nd. Find a corresponding value of τ from Fig. 30, and of κ from one of the Figs. 30_1, 30_2, 30_3, 30_4.

3rd. Calculate $\tan \alpha$ from equation (91_3).

This will not usually correspond to the assumed value of α, but it will show at once in what direction the true value lies. If κ and τ are not changing quickly with the epoch, and α assumed was approximately true, $\tan \alpha$ will be almost exact.

4th. When α has been found, F is determined from equations (90) or (91_2).

Fig. 30.
Scale of epoch.
0° 20° 40° 60° 80° 100° 120° 140° 160° 180° 200° 220° 240° 260° 280° 300° 320° 340° 360°
20° 10° 0 10° 20°
Scale of K.
$\frac{2F}{L_0}$ radius
$\frac{2F}{L_0}$ radius
$n_2=2.00$
$n_2=1.50$
$n_2=1.00$
A
B
C
D
E
F
G
H
K
X
DD=Normal epoch.
Curve BBgives rate of increase of epoch, due to friction, per revolution.
Curve EFGHK gives rate of progression of normal epoch, due to energy from the gear, per revolution.
Curves of Change of Epoch, 72½ Cut Offs $n_2=2.00$
U. S. S. Newark.

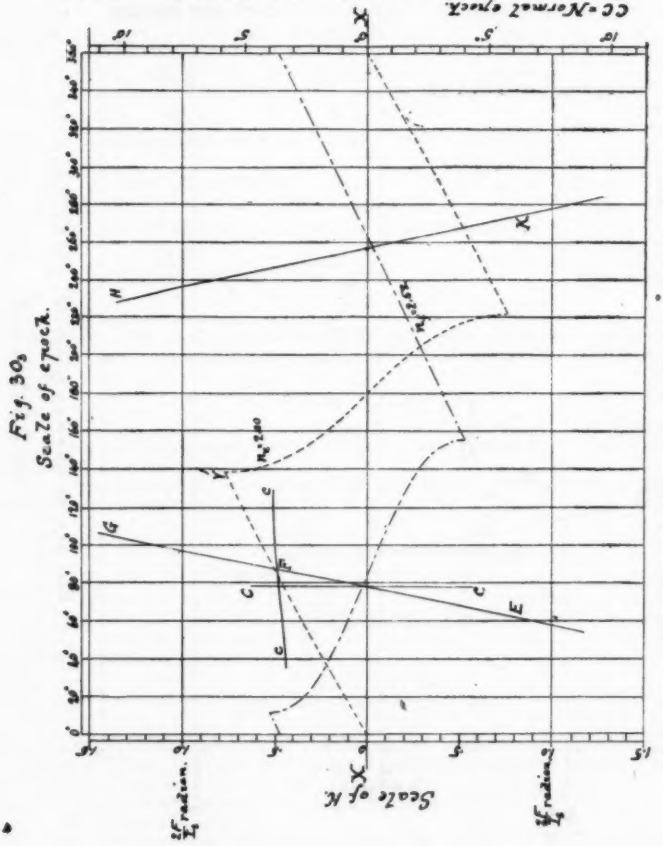

CC = Normal epoch.
Curve CC gives rate of increase of epoch, due to rotation, per revolution.
Curve EFGHK gives rate of restoration of normal epoch, due to energy from the gear, per revolution.
Fig. 30
Scale of epoch.
Scale of K.
Curves of Change of Epoch, .65 Cu. Z Offs n_2 = 2.00.
U. S. S. Newark.

Fig. 30.
Scale of epoch
0° 10° 20° 30° 40° 50° 60° 70° 80° 90° 100° 110° 120° 130° 140° 150° 160° 170° 180° 190° 200° 210° 220° 230° 240° 250° 260° 270° 280° 290° 300° 310° 320° 330° 340° 350° 360°
Scale of X & K_c
$\frac{2F}{L}$ radius
$\frac{4F}{L}$ radius
1.5 1.0 .5 0 .5 1.0 1.5
0 10° 20° 30°
CC = Normal epoch.
Curve ddddd gives rate of increase of epoch, due to friction, per revolution.
Curve EFGHK gives rate of reduction of normal epoch, due to energy from the gear, per revolution.
Curves of Change of Epoch, .65 Cut Off, K_c=1.20
U. S. S. Newark.

As already fully noted, the friction $F = 2163$ lbs., required to keep I_s from rising above $2I_e$ is much larger than the probable value of this quantity, and the total stress of 8.28 tons per square inch is consequently much smaller than would occur if the gear would stand the increased stresses without breakdown. To find the true stress for this cut off at 127 revolutions we would have to repeat the calculations for a lower frictional resistance—say $F = 1400$ lbs.—and find what value of n_e, greater than 2.0, would give this. Obviously the total stress would be much increased.

Probably the first trial of the *Newark*, 10th Dec., 1890, was attempted in this full gear, and it is instructive to learn from *Table IX* that there is no official data. Hence .76 cut off and 127 revolutions are questioned in *Table X*.

.725 cut off, and $n_e = 2.0$.

Here $F = 1463$ lbs. will retain $I_s = 2I_e$ giving a total stress in the studs which broke of 7.61 tons per square inch. F is probably near the truth. We have, then, a stress which is bordering on that at which we would expect a breakdown to occur. See § 70. That this was so is borne out by the Official Report; for, as noted in Table IX, the trial lasted at full power for several hours.

Fig. 30_2 shows that near f, Fig. 29, there would be a triple reversal, and hence the vibration of the gear would be very obvious. One observer speaks in the Official Report of the vibration being "very noticeable."

.65 cut-off.

This is given in *Table IX* as the cut off of the port engine in the 2d and 3d trials. The cut off in the starboard engine in the 3d trial being .62, the stresses would be lower than in *Table X*.

If $n_e = 2.0$, $F = 687$, which would be an exceedingly low value. For a higher value of F, n_e would fall and also the total stresses.

The last column of the table gives the calculation for $n_e = 1.2$. Here, $F = 1240$ lbs.—near the probable value—and the stress falls to 5.15 tons per square inch.

As noted in *Table IX*, before the 3d trial the $1\frac{3}{8}$ inch studs which broke were replaced by $1\frac{1}{4}$ inch studs of Norway iron. But the stresses in the table are calculated for the conditions of the port engine in the last two trials; and we see that even with-

out this change the stresses are within a very safe limit, and the four hours maximum horse power trial was successfully made by both engines.

Figs. 30_3 and 30_4 both still show triple reversals near f, Fig. 29; but, in the latter case, the stress is not much above what would give a single reversal. Probably for this case the gear would appear to run quite smoothly.

Even in the case of .65 cut off and $n_e = 1.2$ it should be carefully observed that the total stress is fully one hundred and sixteen per cent. higher than it would have been estimated had only F and I_f been included, as would usually have been the case.

The estimate of normal pressure in § 66 from which F is estimated arises principally from the outward pressure of the spring rings, only about one-fifth of the total being credited to the weight of the valves. If the rings were allowed to become too much worn, or the scraped surfaces between them and the valve body to get out of order so that the steam would not act efficiently in forcing them out against the face of the liner, *Table X* shows clearly that the greatly reduced friction thus caused would allow I_n, and consequently the total stress, to rise to a dangerous limit if the *Newark* was run at full speed, even at .65 cut off.

73. SHARPNESS OF DETERMINATION OF SAFE ANGLE OF REVERSING LEVER.

This table shows clearly that, notwithstanding the difficulty of determining the safe limit of stress and also the proper value of F—for which no experiments directly bearing on the present case seem to have been made—the extreme rapidity with which the rigid gear amplitude, a_n, for the higher periods, responds to any change in the dimensions of the gear, allows us to determine very closely the limit of the safe angle of the reversing lever. This geat sensitiveness of a_n has been fully brought out in *Tables I* and *III*, and in Figs. 24 and 25. The change of angle of the reversing lever of the *Newark* from .76 to .65 cut off is only $6° - 19'$, as shown by *Table VII*. If the calculations of *Table X* had been made for a constant value of F, as they should have been had the intention been only to bring out the effect of link-

ing up on the total stress, the results would have been much more striking.

74. OFFICIAL REPORTS. ENTIRE AGREEMENT WITH RESULTS IN *TABLE X.*

The following is an extract from the Official Report of the *Newark's* trials:—

"Two unsuccessful trials of the machinery of the *Newark* were made of 10th and 12th December [1890], respectively. In each case the starboard (forward) engine was disabled, by breaking of the bolts which secure the cap brass to the low pressure radial link of the Marshall valve gear. An examination showed that a mistake had been made in the length of the rod connecting the eccentric lever with the low pressure rock shaft arm. This rod was subsequently shortened $\frac{1}{8}$ of an inch, the broken cap bolts renewed and the gear adjusted, after which the angle of vibration in the rock shaft arm was found to be more evenly divided, and the motion of the gear improved.

"On the 22d of December, 1890, the four hours' maximum horse power trial, required by the contract, was successfully made . . . The piston valves and the radial valve gear performed their functions satisfactorily, working noiselessly and without undue friction."

The link *CD*, Fig. 27, being $\frac{1}{8}$ of an inch too long would not sensibly affect the stress on the studs at *B*, in the link *OB*, which broke; as the arms at which the forces, applied at *D* and *G*, act, would be affected in the same way and by only a very small amount. Also the stress comes on these studs when *D* is displaced to the left of the vertical and the longer link *CD* would actually be more favorable, as the levers would be more near the vertical when the studs were under heavy stress. The unequal vibration of the levers *FG* and *FD* would perhaps slightly increase a_n, but the effect must be very small.

Tables IX and *X* very fully explain every particular of the Official Report. The breakdown of only the starboard engine on the second unsuccessful trial is due to the port engine's L.P. valve gear having been linked up. And the fact that the trial lasted several hours at full speed showed that the dangerous

conditions had just been reached. That the greatly improved working and the safety of the gear on the last trial, as noted in the Official Report, could also have been certainly predicted, is brought out by the last two columns of *Table X.*

75. GREAT INCREASE OF a_s' OVER a_s.

The stress arising from the rigid gear synchronizing vibration would be

$$\frac{M\omega_s^2 a_s}{g} = \frac{M\omega^2}{g} s^2 a_s. \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

In the *Newark* from *Table X*, line (l),

$$\frac{M\omega^2}{g} = 12261.$$

Also for .76 cut off, $a_6 = 10^{-3} \times 1.505$ feet.
Hence the rigid gear stress is

$$12261 \times 36 \times 10^{-3} \times 1.505 = 664.3 \text{ lbs.}$$

As I_s given for that cut off is 12,580 lbs., the corresponding magnification of amplitude is

$$\frac{12580}{664.3} = 18.94.$$

The corresponding values for the earlier cut offs are also given in line (oo), *Table* X. As F is near its probable value in the column for .725 cut off and also in that for .65 cut off and $n_s = 1.2$, the corresponding increase a_6'/a_6 in these columns is also the probable one; that is, the magnification in both cases for the *Newark* is about 30 times.

It will still further impress on us the necessity for considering the effect of the synchronizing vibration if we compare these figures with the change of amplitude, a_n'/a_n, which the non-synchronizing vibrations undergo. By equation (22) this is,

$$\frac{1}{1 - \frac{T^2}{T_n^2}}.$$

Equation (21) gives

$$\frac{2\pi}{T_n} = n\omega,$$

and

$$\frac{2\pi}{T} = s\omega.$$

Hence

$$\frac{T^2}{T_n^2} = \frac{n^2}{s^2}.$$

From which

$$\frac{a_n'}{a_n} = \frac{s^2}{s^2 - n^2}. \quad \ldots\ldots\ldots \quad (101)$$

Thus we have for the *Newark*, putting $s = 6$ and $n = 1, 2, 3,$ &c., successively,—

1st period . .	$\frac{a_n'}{a_n} = 1.029$	7th period .	$\frac{a_n'}{a_n} = -2.769$
2d " . .	" $= 1.125$	8th " .	" $= -1.286$
3d " . .	" $= 1.333$	9th " .	" $= -.800$
4th " . .	" $= 1.800$	10th " .	" $= -.5625$
5th " . .	" $= 3.273$	11th " .	" $= -.424$

The negative sign of a_n'/a_n in the 7th and higher periods only indicates that the epoch of these elastic gear vibrations is changed by 180° from that of the corresponding rigid gear vibration.

WATER-TUBE BOILERS IN THE BRITISH NAVY.

The memorandum on this subject recently submitted by the Admiralty to both Houses of Parliament, explanatory of the widespread though unofficial criticisms of the Belleville boiler, as used by the Admiralty so extensively, throws much light upon the subject and brings into the area of illumination such facts as are necessary to a proper judgment of the matter.

The paper is full of interest to the Naval Engineer, and exhibits well the urgent necessity of making engineering knowledge of paramount importance in the personnel of a modern navy. The special merits or demerits of the Belleville boiler have not been fairly presented by the previously published rumors or semi-official reports, and great distortions of the facts have resulted.

While it would be only proper for the Admiralty to make the best presentation possible of the good points of the boiler it has so extensively adopted, the showing brings into a proper prominence the fact that water-tube boilers in general, with the increased rate of evaporation, and higher steam pressures, require a greatly increased skill and intelligence in their management. Both in numbers and in technical training the engineer force of naval vessels unquestionably demands increase. This is slowly, and expensively, dawning upon the higher authorities in the navies of the world. Official reports of facts, when made public as are those in the memorandum, assist in the earlier consummation of a proper distribution of trained men for such service and in advancing the essential knowledge of engineering to the grades of command where it is needed.

MEMORANDUM ON WATER-TUBE BOILERS.

This statement deals only with ships fitted with Belleville boilers or with boilers of large-tube types.

The use of small-tube boilers has been hitherto confined in

Her Majesty's Navy to third-class cruisers and smaller vessels, where the importance of securing a high speed on small dimensions justifies the acceptance of boilers which have a shorter life than is desirable for large ships.

The tubes of small-tube boilers are much thinner than those of the large-tube types, and the maximum power of the boilers is obtained by forcing, which also tends to shorten their life.

The experience gained with these boilers up to the present in destroyers, torpedo gunboats and third-class cruisers does not, in the opinion of the Admiralty, justify their use in larger ships, though they have been adopted for this purpose in France and some other foreign navies.

The Admiralty policy has been to consider the Belleville boiler, with which they have had more experience than any other type of water-tube boiler, as the approved type for large ships, treating other types as experimental until they have shown that they possess some distinct superiority.

There are substantial advantages in adhering as far as possible to one principal type for all ships, as the whole service becomes quicker accustomed to its use, artificers, both in the ships and dockyards, become more expert in its repairs, and there is to a large extent interchangeability of the spare parts and accessories which have to be kept in store.

Two long series of trials have been carried out in the *Sheldrake* and *Seagull*, with the Babcock and Wilcox and Niclausse boilers respectively, for comparison with those previously carried out with Belleville boilers in *Sharpshooter*.

Each of these boilers has its advantages and drawbacks, but the trials showed no decided advantage of either over the Belleville boiler.

The experiments were, however, considered to justify further trials of both these boilers on a larger scale, and it has been decided to fit the Babcock and Wilcox in one of the new sloops, and possibly in one of the two new second-class cruisers, and the Niclausse in a new sloop and in a first-class cruiser of the *Monmouth* type.

This will give a fair trial of these two types of boilers on a suf-

ficient scale in the future, but in this paper it is only necessary to compare the Belleville boiler with the cylindrical boiler.

HIGHFLYER AND *MINERVA* TRIALS.

The trials of the *Highflyer* and *Minerva* form the most direct comparison we have yet been able to make between the performances of Belleville and cylindrical boilers under ordinary sea-going conditions.

The two ships are identical in form and external dimensions, and the principal features of their machinery are described in Table III.

The difference in total weight of the machinery and boilers in the two ships is about 100 tons, which saving of weight in the *Highflyer* has been devoted to giving her 6-inch guns instead of 4.7-inch.

The *Minerva* had been previously employed on relief service to China, and in maneuvers, during which service she had a good deal of trouble with her machinery, but the defects then brought to light had been remedied, and, at the time of the trials, her machinery was in excellent condition, and as she had been some months in commission with a crew turned over from the Training Squadron, her engine-room staff was well trained.

The *Highflyer* had been only recently commissioned for the first time, and, with few exceptions, the crew had had no previous experience with Belleville boilers.

An Inspector of Machinery, who had had considerable experience with Belleville boilers in the Channel Squadron, was sent out in connection with the trials, but was ordered in the first place to remain on board the *Minerva*, so that the management of the machinery in *Highflyer* might be under exactly service conditions.

The following trials were carried out in the order shown, the results being given in the attached Tables I and II.

A Series.—Three runs, each of 60 hours' duration, at 10 knots, which, as regards machinery, were continuous and practically formed one run of 1,800 miles, the boilers being changed for each separate run of 600 miles.

B Series.—Two runs, each of 60 hours' duration, at 14 knots, which, as regards machinery, were continuous and practically formed one run of 1,680 miles.

C_1 *Trial.*—One run of 60 hours' duration, at 17 knots.

C_2 *Trial.*—One run of 38 hours' duration, at 17 knots.

(This trial was then discontinued, owing to defects described in Table II having developed in *Highflyer.*)

The *Highflyer*, after refit, made an experimental run at 10 knots, with the Inspector of Machinery on board, and on completion of this made a further trial of 60 hours' duration under the conditions of the A Series, with a considerably reduced coal consumption (see Table II).

An attempt to complete the C Series was discontinued after 8 hours, owing to hot bearings in *Highflyer*, but after refit this run was completed (see C_2 repeat trial).

D Series.—Two runs of 30 hours' duration at the highest power obtainable.

E Series.—Two runs in which *Highflyer* started with 6 boilers alight out of 18, and remainder not lighted. *Minerva* with 3 boilers alight out of 8, and fires in remaining boilers not lighted. At a given signal the fires were lit as soon as possible in the boilers not in use, and the highest speed possible was maintained for 12 hours from the time of the signal.

F Series.—One run at 10 knots for 60 hours, and then to be increased to full speed at an uncertain time. *Highflyer* with 6 boilers alight and fires laid in remaining boilers. *Minerva* with 3 boilers alight, and fires alight, but banked, in the remaining boilers. After completing $65\frac{1}{3}$ hours the signal was made, and the remaining fires were lit in *Highflyer* and spread in *Minerva*, so as to obtain full power as soon as possible, in order to continue at maximum power for 12 hours, but after $1\frac{48}{60}$ hours *Highflyer* developed defects in one of the air pumps, which caused the trial to be discontinued.

During the earlier trials the coal expenditure in the *Highflyer* was found to be very much in excess of that in the *Minerva*, and so large, compared with the results obtained on her contractor's

trials, as to indicate either serious defects or want of skill in the management of the fires.

The Inspector of Machinery was therefore directed to proceed on board the *Highflyer*, after Trial C_2, and the ship was sent to sea by herself to work independently for the purpose of training the stokers.

After 24 hours of this training, Trial A was repeated, when it was found that the expenditure was reduced from 3.45 pounds per I.H.P. for all purposes, which was the average of the three preceding similar trials, to 3.16 pounds per I.H.P.

The *Minerva's* average expenditure for the three A Trials was 2.96 pounds per I.H.P., showing the excess expenditure of *Highflyer* over *Minerva* at the low speed of 10 knots to be 6.7 per cent.

Before this training was carried out the excess expenditure of *Highflyer* at this speed had been 16.5 per cent. on the average of the three A Trials.

The same difference is shown at 17 knots, if the average expenditure in C_1 and C_2 before the training is compared with C repeat after training. The excess in the former case being 15.9 per cent. and in the latter 6.7 per cent.

The coal expenditure in the *Highflyer* was still so considerably greater than it had been on the contractor's trials, as to indicate that there were other causes of wasteful expenditure which had not been discovered; but the time during which the ships could be spared for the trials was limited, and it was not possible to make any further investigation at the time. During her refit at Devonport it was seen that the low-pressure slides and faces were much worn and not bearing satisfactorily, which would account for considerable loss of economy.

After this refit it was found that her expenditure on passage to the East Indies was, from Plymouth to Gibraltar, at 12.4 knots, 2.74 pounds per I.H.P. for all purposes, and from Gibraltar to Malta, 2.67 pounds, at 13.16 knots, which shows a considerable improvement.

Just previously to this the *Diana*, a similar ship with cylindrical boilers, made the same passage in the opposite direction

while returning from Australia. The speed and coal expenditure of the two ships compare as follows:

	Malta to Gibraltar, or *vice versâ*.				Plymouth to Gibraltar, or *vice versâ*.			
Ships.	Distance in miles.	Speed.	Coal per I.H.P.	Total coal used.	Distance in miles.	Speed.	Coal per I.H.P.	Total Coal used.
		Knots.		*Tons.*		*Knots.*		*Tons.*
Diana	1,023	12.0	2.94	207	1,072	12.6	2.85	235
Highflyer.....	1,023	13.16	2.67	191	1,072	12.4	2.74	215

Diana's port engine was stopped for five hours between Gibraltar and Plymouth.

The most important point, however, which had to be determined by the trials was the maximum speed that could be maintained at sea under ordinary service conditions. It is, therefore, the trials at this maximum speed which are most important.

In the design of machinery for a man-of-war the main problem is, how to combine certain conditions of armament, protection, coal endurance, etc., with sufficient speed to catch the enemy's ships when they are sighted at sea.

Trials D_1 and D_2 at the highest speed obtainable for 30 hours show an advantage in speed of .37 of a knot in the first, and .94 of a knot in the second, in the favor of *Highflyer* and this was confirmed by Trials E and F.

The I.H.P. exerted by *Highflyer* was 396 more than *Minerva* in D_1 and 1,382 in the other trial; and if *Minerva's* machinery were to be redesigned to give this increased power retaining cylindrical boilers, it would have to be from 47 tons to 170 tons heavier.

Moreover, the boilers and machinery would require more space, and this would require a larger ship with increased weight of hull, which would again necessitate more I.H.P., and consequently a further increase of weight to maintain the same speed. This would, of course, be in addition to the 100 tons already gained in *Highflyer*.

The Trials E and F, which were intended to show the advan-

TABLE I.—H. M. S. *MINERVA*—RESULTS OF SPECIAL TRIALS.

Description of trial.	Date.	Duration of trial.	I.H.P.	Speed.	Revolutions.	Draught of water. Forward.		Draught of water. Aft.		Number of boilers in use.	Coal consumption per I.H.P. per hour, all purposes.	Water. Used from reserve tank.	Water. Distilled for machinery.	Water. Distilled for ship.
		Hours.		*Knots.*		*Ft.*	*Ins.*	*Ft.*	*Ins.*		*Lbs.*	*Tons.*	*Tons.*	*Tons.*
A_1	1– 3, 1, 1900	60	1083.8	9.84	66.5	19	10	23	7	3 and 4	2.88	...	5	15
A_2	3– 6, 1, 1900	60	1084.5	9.89	66.19	19	6	23	4	4	3.02	...	5	15
A_3	6– 9, 1, 1900	60	1084.95	9.98	65.98	19	1	23	1	4	2.99	...	5	15
B_1	13–16, 1, 1900	60	3028.1	14.00	93.47	19	11	23	5	6	2.33	...	7·5	23
B_2	16–18, 1, 1900	60	3169.0	13.93	94.2	18	11	23	2	7	2.24	...	10	22
C_1	24–26, 1, 1900	60	6242.35	17.15	117.92	18	9	23	1	8	2.28	...	20	6
C_2	30, 1, 1900 to 1, 2, 1900	38	6022.8	17.05	117.1	19	0	23	3	8	2.25	...	9	...
C_2 repeat.	28, 2, 1900 to 3, 3, 1900	60	6021.7	17.0	117.3	19	2	23	1	8	2.22	...	15	...
D_1	6– 7, 3, 1900	30	7550.2	17.53	123.9	19	3	23	2	8	2.23	20	24	...
D_2	10–11, 3, 1900	30	7379.2	18.03	123.6	18	2	22	11	8	2.16	...	12	...
E_1	17, 3, 1900	$13\frac{30}{60}$	6778.16	16.98	119.8	19	6	23	3	8	2.16	...	2	...
E_2	20, 3, 1900	$13\frac{30}{60}$	6510.3	16.83	117.3	19	0	23	0	8	2.11	...	2	...
F	24–27, 3, 1900	$65\frac{20}{60}$	1132.0	10.0	66.1	19	10	23	3	3 to 8	3.09	...	5	...

C_1 S.H.P. cylinder cover leaked badly.
C_2 S L.P. slide-rod gland leaked badly. Trial curtailed on account of defects in *Highflyer*.
C_2 No defects. This trial is a repeat of two former trials.
D_1 Forward main feed suction pipe burst. S.H.P. piston-rod gland leaked badly.
D_2 Starboard main feed suction pipe burst.
F No defects. Trial at full power discontinued at end of $1\frac{30}{60}$ hours on account of defects in *Highflyer*.

TABLE II.—H. M. S. *HIGHFLYER*—RESULTS OF SPECIAL TRIALS.

Description of trial.	Date.	Duration of trial.	I.H.P.	Speed.	Revolutions.	Draught of water. Forward.		Draught of water. Aft.		Number of boilers in use.	Coal consumption per I.H.P. per hour, all purposes.	Water. Used from reserve tanks.	Water. Distilled for machinery.	Water. Distilled for ship.
		Hours.		*Knots.*		*Ft.*	*Ins.*	*Ft.*	*Ins*		*Lbs.*	*Tons.*	*Tons.*	*Tons.*
A_1	1– 3, 1, 1900	60	1190.5	9.8	86.35	*19	8	22	11	6	3.46	...	42	19.5
A_2	3– 6, 1, 1900	60	1205.5	9.9	85 9	to		to		7	3.38	...	46	12
A_3	6– 9, 1, 1900	60	1137.4	9.98	85.85	†18	9	21	6	8‡	3.50	...	48	19
A repeat.	14–17, 2, 1900	60	1101.9	9.82	85.89	20	0	22	2	5	3.16	...	16	14
B_1	13–16, 1, 1900	60	2831.5	14.0	120.9	*20	6	22	5	12	2.90	...	44	10.5
B_2	16–18, 1, 1900	60	3132.4	13.9	122.75	†18	3	21	7	12	2.87	6	68	21.0
C_1	24–26, 1, 1900	60	5750.4	17.15	153.75	19	2	22	2	18	2.67	16	52	6.0
C_2	30, 1, 1900 to 1, 2, 1900	38	5707.8	17.05	152.6	19	11	22	6	18	2.58	14	44	...
C_2 repeat.	28, 2, 1900 to 3, 3, 1900	60	5907.0	17.0	152.05	20	0	22	6	15	2.37	...	45	...
D_1	6– 7, 3, 1900	30	7946.3	17.9	164.35	19	5	22	6	18	2.52	10	50	...
D_2	10–11, 3, 1900	30	8760.9	18.97	170.48	18	0	21	11	18	2.58	43	50	...
E_1	17, 3, 1900	12$\frac{45}{60}$	9132.1	18.08	171.2	19	10	22	6	18	2.36	10	12	...
E_2	20, 3, 1900	12$\frac{25}{60}$	9268.2	18.71	171.55	18	8	22	4	18	2.47	10	12	...
F	24–27, 3, 1900	65$\frac{20}{60}$	1235.7	9.99	85.9	19	10	22	6	6	3.06	...	40	...

NOTE.—On 9, 2, 00 after steaming 24 hours at 10 knots for the purpose of training the crew, the shafts of both auxiliary circulating pumps broke.
* Draught at commencement of trials. † Draught at end of trials. ‡ Decreased to 6.

A_1 Two generator-door joints and seven economizer-door joints leaky. Two fusible plugs renewed. These defects caused one boiler to be out of use 4¾ hours.
A_2 Two economizer-door joints leaky and four fusible plugs renewed. These leaky doors caused one boiler to be shut off and another substituted.
A_3 Joints of two doors leaky causing two boilers to be shut off.
A No defects developed. This trial was a repetition of the three previous trials.
B_1 Starboard evaporator feed-pump rod and crosshead defective.
B_2 Ten joints in main steam pipe and many smaller joints leaky. Twelve fusible plugs renewed.
C_1 Crank shaft of P. distilling engine was broken on two occasions. S. crank-pit pump, connecting and pump rods broken and barrel cracked. H.P. cylinders, three stays for supporting cover found broken after trial.
C_2 Crank shafts of both distilling engines broken. S.I.P. crankhead bearing heated, causing curtailment of trial, and spare brasses were fitted.
C_2 End door of steam drum on No. 13 boiler leaky. Top of float box of No. 18 boiler defective Several joints in main and auxiliary steam pipes leaky.
D_1 Impeller shaft of S. auxiliary circulating pump broken. A few joints in steam pipes leaky. Sixteen fusible plugs renewed.
D_2 Two joints in main steam pipe leaky. Impeller of Sd. auxiliary circulating pump worked loose on shaft. Thirty fusible plugs renewed. On subsequent examination junk-ring keep of P.A.L.P. piston was found broken.
E_2 P.I.P. crankhead bearing heated, spare brasses fitted. One fusible plug renewed.
E_2 P.H.P. and P.I.P. crank brasses worked warm. One fusible plug renewed.
F Bucket of P. air pump loose on rod causing damage to webs of bucket and guards of valves. This caused the full-power trial to be discontinued after 12$\frac{45}{60}$ hours.

tage of the water-tube boiler in rapidly raising steam, were largely in favor of the *Highflyer*, except as regards the accident which stopped Trial F. This is a point only second in importance to that of maintaining the maximum speed. It is sometimes said that a captain in war will never let his fires out, but there will be many cases in which he must do so, both for the sake of the overhaul, cleaning, and repair of his machinery and boilers, and also when it is of vital importance to husband his supplies of coal.

After her return to Devonport the work described in the Appendix, No. 1, was done to the *Highflyer*. A complete list of the work is given, as it is a fair example of the kind of work that is found necessary in Belleville-boiler ships commissioned for the first time, after they have had sufficient running to find out their defects.

It will be seen how small a proportion has anything to do with the boilers.

TABLE III.—PARTICULARS OF MACHINERY OF *MINERVA* AND *HIGHFLYER*.

Name of vessel.	*Minerva.*	*Highflyer.*
Engines:		
Type	Triple expansion (3 cylinders).	Triple expansion (4 cylinders).
I.H.P. maximum	9,600	10,000
Revolutions	140	180
Stroke	3′ 3″	2′ 6″
Boilers:		
Type and number	Tank, 8 in No.	Belleville, 18 in No.
Steam pressure	155 lbs. at boilers, 150 lbs. at engines.	300 lbs. at boilers, 250 lbs at engines.
Weight:		
Main and auxiliary machinery.	380.1 tons.	373.8 tons.
Boilers	557.8 tons.	461.5 tons.
Total (ready for steaming)	937.9 tons.	835.3 tons.

TRIALS OF *HERMES* AND *DIANA*.

Another series of trials of a different character was made between two other ships of the same classes as *Highflyer* and *Minerva* respectively, with the object of separating the performance of the boilers from that of the machinery.

E. P.H.P. and P.I.P. crank brasses, spare brasses fitted. One fusible plug renewed.
F. Bucket of P. air pump loose on rod causing damage to webs of bucket and guards of valves. This caused the full-power trial to be discontinued after 88 hours.

The results of these trials are given in Tables IV and V.

It is impossible under ordinary conditions of steaming to separate the waste of heat due to the boiler from the waste of heat and steam that occurs in all parts of the main and auxiliary machinery.

In order to separate the performance of the boilers from that of the machinery, the *Hermes* (whose machinery is identical with that of *Highflyer*) and *Diana* were fitted with tanks for measuring the quantity of water passing through the main and auxiliary condensers.

At the higher powers the quantity of water passing through the engines, and the coal consumption, were both less in *Hermes* than in the *Diana*, whereas at the lower powers, although the coal consumption was slightly less in *Hermes* than in *Diana*, the water used by the former was more, showing that the boiler was more economical, but more steam was used in the engines.

Higher pressures necessarily require greater perfection in all such parts as slide valves, piston rings and stuffing boxes, as well as in all steam joints, as the higher the pressure the more steam will pass through any leak without doing useful work. The waste of steam from stuffing boxes and joints is generally visible, but that which passes through any defective piston rings and slide valves, in either the main or auxiliary machinery, can generally only be discovered by taking the machinery to pieces.

There is no doubt that economy largely depends on the stoking, and on the boiler being free from deposits, and in all respects in good order, but in many cases when the coal consumption has been large, it has been found to be mainly due to defects in the machinery, which, under the conditions of naval service, are very difficult to discover, as Her Majesty's ships are always kept, as far as possible, ready for immediate service, and opportunities for opening up the machinery only occur at comparatively long intervals.

GENERAL OBSERVATIONS ON DEFECTS.

The adoption of water-tube boilers carried with it a great increase of steam pressure, namely, from 150 pounds to 250 pounds at the engines, and this involved alterations of detail in

the designs of practically all the auxiliary machinery, as well as the main engines. It is in these details, and in the precautions necessary for dealing with the higher pressures, that the principal difficulties have occurred.

The principal defects from which the ships have suffered have been:

Leaky Joints.—These occur principally in the early part of a ship's commission. A better method of making them is being introduced as they give out, and all serious trouble from this cause is generally got over after a ship has been some time in commission.

Leaks in Condensers.—This is a defect that is not confined to water-tube boiler ships, and is not affected by the use of high-pressure steam, being principally caused by the splitting of condenser tubes. In new large ships the condensers are now being made into two parts, and in order that one-half may be overhauled and cleaned or repaired while the other is in use.

Evaporators and their Accessories.—Leaky joints and other causes of waste of water have in many cases thrown too much work on these, but these difficulties rapidly disappear when the causes of abnormal waste are removed, and the evaporating plant is now being increased in all modern ships.

Auxiliary Circulating Pumps.—Troubles have been experienced owing to the frequent fracture of the shafts of these engines; generally the fracture took place in the impeller shaft, but in addition to this in the *Canopus*, the crank shaft also fractured, and so also did the engine bed. In the case of the *Canopus*, entirely new engines have been fitted. The fractures of the impeller shafts in this and some other ships were largely due to defective material, although they were also of rather too small diameter for the work required of them. In these cases they have been replaced by the contractors under their guarantee.

Auxiliary Air Pumps.—These have given trouble in some cases, owing to want of strength in the pumps or details of engines. In others, the cause has been the inferior arrangement of the suction or discharge pipes, principally the latter. Stronger forms of double-crank pump are being fitted for this purpose for the

newer ships. In the ships where this defect occurred it has been remedied, either by fitting new pumps or making certain modifications in the existing pumps. This latter course was adopted in the *Goliath*, in which ship the connecting-rod bolts broke, the engine seats were not stiff enough, and the engines and pumps noisy, and with a tendency to wear rapidly. The pump was altered from the ram to bucket type, lighter valves fitted, and the foot valves were omitted, the engine seat stiffened up, and the lead of discharge pipe improved, after which they worked more satisfactorily.

Piston-rod Glands Leaking.—These have given trouble in some cases, especially in the earlier ships. In some cases it was probably due to the design not being suitable to the pressure. The principal cause is either want of adjustment of the packing, or dirt obtaining access to the cylinder and cutting the working parts of the packing or rods. In the *Furious*, considerable leakage of high-pressure glands was due to the scale, from the boilers or steel castings, being carried over by the steam and cutting the packing and rod.

Feed Pumps.—These are all direct pumps without crank shafts. In one variety the sticking of the shuttle valve has often given trouble, causing the shuttle valve to work sluggishly, so that the engine is difficult to start, and may slow down or stop. The cause is generally the presence of dirt, or if salt water is present in the boilers a deposit of salt is often left on these valves. An efficient lubricator is now fitted to them. The joints of the water and steam cylinders have also given trouble.

In the pumps of another design severe corrosion has been experienced in the slide valves and liners, causing these parts to become grooved and leak, so that the engine would not work properly. This was remedied by fitting forged-bronze valves and liners, since which there has been no further trouble.

In the *Hermes* the pumps failed owing to their not having been efficiently overhauled and their defects ascertained and remedied.

Generally speaking, the feed pumps of our ships work well, but as in the new vessels they had to work at much higher pres-

sure, leaks and other defects which would be of little importance with low pressure assume greater importance and cannot be neglected.

Drain Traps and Drain System.—The system of drain pipes, drain traps, drain tanks and drain-tank pumps fitted for dealing with water, in the steam and exhaust pipes, has given trouble in several ships. Defects are principally due to leaky joints in drain pipes, burst drain pipes, defective drain valves or cocks, defective drain traps which fail to work either by becoming choked up with dirt or by allowing steam to blow straight through. Serious losses of water and steam occur from failures in this system, also the temperature of water in the drain tank becomes excessive, and the drain-tank pump gives trouble, and fails to pump out this water.

Considerable attention has been devoted to the design of an efficient steam trap, but it is doubtful whether any of them are so at present, and if *efficient* when *new* they soon become deranged. In some recent ships considerable simplification of the drain system has been made, the drains for steam pipes, for instance, being fitted as simple pipes discharging into the bilge without being connected to any steam trap. This will involve more care being taken in raising steam, but will, no doubt, result in the saving of much annoyance, and prevent much loss of steam, which seems to be inevitable in the old system.

Main Engine Slide Valves.—Where these have been of the flat type with relief rings at their back, trouble has been experienced in some ships owing to the inefficient action of the relief ring. They require very careful adjustment and great care taken to ensure that the rings are capable of moving freely in their recesses. In the case of flat slide valves to intermediate-pressure cylinders, their failure has often been accompanied by cutting of valves and faces. Flat slide valves have now been abandoned in the newer vessels for the intermediate cylinder, and valves of the piston type are fitted, these being free from this source of trouble. Cutting of faces of the flat slide valves on L.P. cylinders have also been experienced, but this has been generally due to the use of too soft cast iron.

In the boilers themselves the following defects have been the most common :—

Corrosion of the Baffle Plates in Steam Collectors.—These, being made of thin plates, corrode rapidly, and the rust and scale are liable to be carried into valve seatings or other places where they may do harm, though the greater part of the scale is deposited in the mud drums. Modifications have been made in these fittings in *Terrible*, *Diadem*, *Furious* and *Arrogant*, which it is expected will greatly reduce the amount of corrosion. New ships are being fitted similarly, and completed ships altered as opportunities offer.

Joints of Doors Leaking.—This occurred to a serious extent in the *Argonaut* during last year's maneuvers, due to the doors not being interchangeable and to unsuitable asbestos washers being used for making the joints.

Since suitable washers have been provided very little fresh trouble has arisen from this cause.

Float Gear of Automatic Feed Corroding.—Portions of these specially liable to corrosion have either been galvanized or bushed with bronze as opportunities occur, and there has been very little trouble with them lately.

Non-return Valve in Down Comers.—The pins of these became corroded in the *Terrible*, and the valves failed to act, but since attention has been called to them and they have been refitted, no fresh trouble has arisen.

Supports for Back Ends of Tubes Getting Hot or Burning Away.—Such heating has happened in some cases, where they were insufficiently protected by the brickwork, or where the latter has been destroyed. For new boilers, and also in cases where old brickwork becomes defective, it is built up in an improved manner, and its behavior is then satisfactory.

Nickel Washers Leaking.—This is generally caused either by bending of the generator tubes, or the dropping of the back ends by the burning away of the supports at the back of the tubes, as above. When the supports are properly protected, and the tubes are not allowed to get overheated from imperfect circulation or

shortness of water, leakage rarely occurs. In any case the washers can be very quickly duplicated or replaced.

Pitting of Tubes.—This occurred to a considerable extent in the *Diadem*, 13 tubes in all being pitted through. Methods of preventing it are under trial, namely, galvanizing the interior of the tubes, and the use of zinc rods; also more care is being taken to prevent the water from becoming acid, by the use of lime.

Leaky Safety Valves.—These are a frequent source of trouble, which is increased with the higher pressures of steam. Leaving out the greater difficulty of making tight joints at high pressures, in many ships the cause has been the cutting of the valves and seats by scale and other foreign matter blown out of the steam collecter. Where this has occurred, it has been remedied by having the steam collectors thoroughly washed out and the valves refitted.

With the newer machinery designs the simplification and galvanizing of the internal baffle plates will reduce the liability of leakage due to this cause, and as opportunities offer the steam collectors of the older designs are also being galvanized and simplified.

In a few cases, the leaky safety valves experienced were due to defective designs, which were not suited to the high pressure, and new valves were fitted.

Bursting and Overheating of Belleville Boiler Tubes.—Either bursting or overheating has occurred in the *Powerful*, *Terrible*, *Arrogant*, *Furious* and *Hermes*. In new ships, on contractor's trials, when overheating has occurred, it has been principally through obstructions to the circulation of water by some foreign matter, such as wooden plugs or waste being left in the boilers. In a few cases, defects of manufacture have been the cause, especially before the manufacture of solid-drawn tubes was properly developed. This was the case in the *Powerful*. In ships in commission the cause has generally been either shortness of water or the presence of saline deposits in the tubes due to the admission of sea water, while the necessary blowing off, to keep down the density under these conditions, was neglected. This was the case in the *Terrible*. In the *Arrogant* there was a combination

of these two causes. In the *Furious* only two very small splits occurred in two adjacent tubes. They were not observed until the boilers were examined after steaming. The tubes were clean, but the general appearance of the elements of which these tubes formed part showed that overheating was the cause. No doubt this boiler had been short of water.

With tubes of the quality now supplied to our boilers, provided solid deposits are excluded, and the proper quantity of water is kept in the boilers, this source of trouble should be eliminated.

Salt-Water Obtaining Access to Belleville Boilers.—Cases have occurred in which sea water has been used in these boilers without much inconvenience, provided proper precautions are taken, but on the return voyage of the *Terrible* from Malta with relief crews the undesirability of allowing any access of sea water was shown. Although, as stated above, sea water can be used if care and due precautions are exercised, yet the small quantity of water contained in the boilers renders the rise of density comparatively rapid, and the danger of saline deposits correspondingly great, especially at low rates of combustion. In the *Terrible* many of the tubes were salted up and burst, or were otherwise damaged. The sea suctions of all feed pumps have now been blanked off to prevent the unauthorized use of salt water, this and leaky condensers being the usual causes of salt water getting into the boilers.

The total number of tubes that have been removed as defective from each ship is given in Table No. XIII.

It will be seen from this table that, excluding *Hermes*, which is an entirely exceptional case, most of the defective tubes occurred in the *Powerful*, *Terrible* and *Arrogant*, which were the first ships commissioned. The greater part of these occurred in the earlier part of their commissions, when the management of the boilers was less understood.

Since the accident to the *Terrible*, in the previous paragraph the number of tubes damaged has been very much reduced.

Table No. XIV shows the total number that have become de-

fective in the whole fleet since the date of that accident, excluding *Hermes.*

Boiler Tubes Being Pulled Out of Junction Boxes.—This has occurred in two ships, the makers of the boilers being the same in each case. It was due to the thread in the junction box being defective, and was a case of bad workmanship which the overseers had not detected.

Feed-Collector Tubes Laminated.—In a few cases laminations have developed in these tubes, principally on the fire side. The cause was defective manufacture.

Where this defect has occurred, however, it has not been such as to interfere with the steaming of the particular boiler. The laminated tubes have been removed and new ones fitted. These feed collectors are now being largely made from solid bars bored hollow.

Smoke-Box Doors of Belleville Boilers Defective.—These doors are found sometimes to warp and buckle under the influence of excessive heat, occasionally resulting in the bolts of the doors not entering properly into their sockets. This defect, if not remedied, causes a loss of economy in combustion. The overheating is generally due to inferior stoking. In the newer ships the smoke-box doors are being made of smaller dimensions and of a stiffer design, which will reduce the tendency to this defect.

BEHAVIOR OF BELLEVILLE-BOILER SHIPS IN COMMISSION.

Tables of Trials while in Commission.—Tables VI to XII* have been prepared to show the practical results of the working of the machinery of ships fitted with Belleville boilers on various passages and trials while in commission, the cylindrical-boiler ships being printed in italics:

Arrogant and *Furious* had a considerable advantage in speed but a much larger coal expenditure.

The ships of this class being purposely made short, in order to be used as rams, require considerably more H.P to drive them at the same speed than the *Dido* class, and their consumption per I.H.P. was also high.

These ships are not fitted with economizers.

* Tables VI, VII, X, XII are omitted in this reprint as least important.

Table VIII similarly compares *Diadem's* run from Gibraltar to Sheerness, and *Andromeda's* run from Gibraltar to Malta, with the performances of *Blenheim*, *Blake* and *Royal Arthur*, and with the celebrated passage of the United States cruiser *Columbia* across the Atlantic, which are the best long distance passages of cruisers with cylindrical boilers of which we have records.

Table IX shows the trials at high speed made by *Powerful* and *Terrible*, while in commission on the China Station and in the Channel respectively.

It will be seen that the *Ocean*, with Belleville boilers, had an advantage in speed over all the other battleships; the *Renown*, which was the next ship, being .87 of a knot slower.

Table XI gives the ordinary Quarterly Passage Trials of ships with Belleville boilers not included in the other tables, with a trial of *Niobe* for 60 hours at 3/5 power.

In comparing the total coal expenditure, allowance must be made for the difference of displacement and the speed at which the voyage was made by the respective vessels.

Powerful and Terrible.—The *Powerful*, which was commissioned in June, 1897, after serving in the summer maneuvers of that year, left for China in the following October. Only minor defects occurred on the passage, as far as Mauritius. None of them were of sufficient importance to cause any delay or stoppage of the engines.

After leaving Mauritius she was ordered to make a trial at half power, which resulted in some of the crank-head bearings getting hot, and damage to the bearings of the port engine, which made it necessary to stop that engine for 2½ days. The distilling pumps were also damaged, and these were repaired at Colombo.

On arrival at Hong Kong the main bearings had to be readjusted, and numerous minor defects remedied, but no large repairs were required to the boilers.

After these repairs the ship remained generally efficient during the whole of her commission, though 20 downcomer tubes, 34 generator tubes and four complete elements had to be renewed at various times.

Her various high speed trials and passages are given in Table IX.

The high-speed trials of *Terrible* are also given in the above table, and this ship also made a passage from Portsmouth to Malta, a distance of 2,200 miles, without stopping, in 1898, at an average speed of 17.7 knots.

The *Terrible* had a serious accident in March, 1899, the bursting of one of the tubes while the furnace door was open, causing the death of a stoker and injuries to four other men.

The cause of this accident was attributed, after careful inquiry, to the unauthorized use of salt water for feed.

On arrival at Portsmouth after this accident, the boilers were thoroughly overhauled, and a number of tubes renewed, namely, 78 generator tubes and seven complete elements. The baffle plates in all the steam collectors which in the earlier Belleville boilers rapidly corrode, were removed and replaced by a simpler arrangement, which was galvanized.

Since leaving England for South Africa and China, no defects of any importance have been reported. Her coal consumption on the passage to the Cape was, however, very large. She is now at Taku, and is being kept by the Admiral fully manned and ready for any emergency.

Channel Squadron.—The *Diadem*, *Niobe*, *Arrogant* and *Furious* were serving in the Channel Squadron from the respective dates of their commissioning in 1898 until the outbreak of the South African war, when they were despatched on various services connected with the war.

They have been generally efficient, though the coal expenditure of *Arrogant* and *Furious*, as of all the ships without economizers, has been high.

The advantage of quick raising steam was illustrated in the case of the *Niobe* at Las Palmas, when the *Persia* transport broke her shaft and was nearly on the rocks. The *Niobe* was able to get up steam and go to her assistance in 1¾ hours from the time of receiving the news. With cylindrical boilers it would have required five or six hours.

The *Arrogant* on one occasion ran short of coal on the voyage from Gibraltar to Plymouth, and was towed for 24 hours by the *Magnificent* as a precaution, but arrived at Plymouth with 115

tons of coal on board. This was mainly due to her having been supplied with a very bad quality of coal, which was received during the coal strike.

In *Furious* the baffle plates in steam collectors were altered on a similar plan to those in *Terrible*, and in *Diadem* and *Arrogant* this alteration was partially carried out.

While at Las Palmas a few of the tubes in *Diadem's* boilers were found to be pitted through and were plugged, and on her arrival in England eight generator and five economizer tubes were renewed on this account.

One economizer and one generator element were also removed, to be cut up for examination, and one economizer tube in each of six boilers was renewed for comparison with the new galvanized elements. Zinc rods were also put in the top tubes for experiment.

During the commission, *Furious* has had six generator tubes renewed out of a total of 2,880, and *Arrogant* 15 generator tubes and one element complete out of the same number.

Niobe has had one tube renewed.

Naval Maneuvers, 1899.—In these maneuvers there were five Belleville-boiler ships in A Fleet, which had all but one been some time in commission, and five in B Fleet, which were all commissioned for the maneuvers.

No complaints were made of those in A Fleet. Three of them were selected for the routes that required the highest speed in the plan of search, namely, *Niobe*, 18 knots, and *Arrogant* and *Furious*, 17 knots.

In the B Fleet, during the preliminary cruise, the *Argonaut* developed serious leaks from her boiler doors, as well as numerous leaky joints in steam pipes, and 31 artificers from the fleet were sent on board at Milford to repair them. The leaks in the boilers were mainly attributed to the adoption of unsuitable asbestos washers for making the joints. This has been remedied, and very little trouble is anticipated from this cause in future.

The *Europa* suffered in a minor degree from the same cause, but not to the same extent; the leaks were in steam joints and

not in the boilers. She made good her defects with her own resources.

All four Belleville-boiler ships, *Europa*, *Argonaut*, *Gladiator* and *Vindictive*, with *St. George*, *Diana* and *Sappho*, were given the task of picking up the convoy, and maintained a speed of 17 knots, except during fogs, until the convoy was reached. *Sappho*, however, one of the cylindrical-boiler ships, could not maintain this speed, and was sent back.

Afterwards, the four Belleville-boiler ships were stationed off Cape Clear, where they came in touch with the enemy's fleet, and chased and captured *Furious* and *Pactolus*. *Europa* during this time had a hot bearing which was rectified in time to enable her to join in the chase, and *Argonaut*, running short of water, was obligated to return to Milford.

Of the cylindrical-boiler ships of B Fleet, the *Rainbow* ran short of water, and the *Sirius* was unable to do more than 13 knots.

Ships Commissioned since the Maneuvers.—Of the ships commissioned since the maneuvers, the *Hermes* is the only seriously bad case of a breakdown. This case formed the subject of a court of inquiry, the minutes of which are still under consideration.

The cause of failure was great leakage of water, combined with persistent failures of the feed pumps. Certain parts of these pumps were examined on several occasions, and the defects which were discovered remedied, but the principal defects appear to have escaped observation, with the result that on passage from Jamaica to Bermuda the ship had to be stopped at sea for three days while temporary repairs were carried out, and afterward to put into Nassau.

She was afterwards towed to Jamaica, where temporary repairs were carried out, and then proceeded to Bermuda, partly under her own steam and partly in tow of *Tribune*. The boilers were seriously damaged, and a large number of tubes will have to be replaced.

This ship is similar in all respects to the *Highflyer*, which has been through a long series of trials with *Minerva*, and the machinery is by the same makers.

The *Highflyer* had numerous small defects, but nothing of a serious nature, and nothing at all resembling those which occurred in *Hermes*. On her voyage to the East Indies, the behavior of the machinery as far as Malta has been entirely satisfactory.

The *Europa* has been employed in relief service. Her coal expenditure has been abnormally large, and an Inspector of Machinery was ordered to go in her on the voyage to Australia in which she is now employed, with a view to discovering the cause. This he has not yet succeeding in doing. The ship is now on her return voyage to England, and the matter will be further investigated on her arrival.

Her sister ship, the *Argonaut*, has just left England for China, and used very little more coal on the voyage to Gibraltar at 14.5 knots than the *Europa* did at 11.7.

Canopus, *Ocean* and *Andromeda*, have had no serious defects since they were commissioned. *Canopus* has had minor troubles with her auxiliary air and circulating pumps, which are now being replaced by new ones supplied by the contractors.

In order to give a better opportunity for training the engine-room staff and of discovering defects before the ships leave their ports, it was decided in the case of *Goliath* and *Argonaut* to make a series of trials of more prolonged duration than usual on commissioning, returning to port after each trial to remedy any defects that were discovered.

This course involved a certain loss of time in getting the ships away from England, but it is only reasonable that when the machinery involves so much that is new, time should be given both for training the crew and for more thorough tests than would otherwise be necessary.

A large number of minor defects were discovered. Both ships have now left for China. The *Goliath* has reached Colombo with very moderate coal consumption, where it is compared with that of *Victorious*. The behavior of the engines has been reported satisfactory, except for the breakdown of one auxiliary air pump between Aden and Colombo.

The *Argonaut* made the voyage to Gibraltar at a speed of 14½

knots with moderate coal expenditure, namely, 2.3 per I.H.P. for all purposes.

THE ADOPTION OF WATER-TUBE BOILERS IN FOREIGN NAVIES.

Practically all foreign nations may say to have adopted water-tube boilers in some form or another, almost to the entire exclusion of cylindrical boilers.

France was the first in the field, and has adopted many different types.

The principal ones are the Belleville, Lagrafel D'Allest, Niclausse, Normand, Guyot, and Du Temple boilers, the last three being small-tube types.

Germany has in some cases combined cylindrical with water-tube boilers of small-tube types, either Thornycroft's or Schültz, which is similar. They have also the Dürr, Niclausse and Belleville.

Russia has principally Belleville boilers, with a few Niclausse, Schültz, Normand, and other kinds.

United States have combined cylindrical with Babcock and Wilcox or other types in some of their ships. Their new vessels are all to have water-tube boilers. Two battleships under construction have Thornycroft boilers and one Niclausse.

Italians have Belleville or Niclausse in their large ships building.

Dutch have Yarrow, or Yarrow in combination with cylindrical boilers.

Norway and Sweden principally Yarrow.

Japan, Belleville.

A list giving the types of water-tube boilers fitted and to be fitted in the war vessels of the principal foreign navies above third-class cruisers is given in Appendix No. 2.

CONCLUSION.

Surprise is naturally felt by those who are not fully coversant with the whole of the circumstances, that difficulties connected with the management of water-tube boilers and high-pressure machinery should take so long to overcome, and they are apt

to assume that because they are not all removed in the three or four years that have elapsed since water-tube boilers were introduced into the British Navy, they must be insuperable.

A little consideration will show that up to the present, and for some time to come, the engine-room staff of every newly-commissioned water-tube boiler ship must be largely composed of those who have had no previous experience of this type of machinery, as the number of water-tube boiler ships in commission has up to the present borne so small a proportion to the total number of ships for which crews are provided. The rate at which crews can be trained will increase rapidly as more water-tube boiler ships become available, and as special arrangements for training engineers, engine-room artificers and stokers augment the number of men with experience of these boilers.

When defects of any importance occur in any part of the machinery of new ships, the best method of dealing with them is considered by the makers of the machinery, who are responsible for the design and have great interest in maintaining its efficiency; by the dockyard officers, who have the experience of all similar defects that have been dealt with at their own yard, and by the Admiralty engineers, who are in close touch with all the dockyards, as well as with all the contractors and the officers afloat.

For each defect, the first thing is to determine the true cause, which is often difficult to ascertain. It must then be determined whether the defect might not be avoided if some different method of treatment was adopted, or some precaution to prevent improper management was introduced, and, finally, if an alteration is to be made, what the alteration should be in each case. Each of these steps requires time for investigation and experiment, which must, as a rule, be carried out without interfering more than can possibly be helped with the service on which the ship is employed.

Alterations, when decided on, can only be carried out when the ship can be spared, which is generally only when she is in dockyard hands for other purposes, and the efficiency of each

alteration must generally be tested by actual experience in one or two ships before it is carried out generally.

Improvement is, however, steady and continuous, and may be expected to become more rapid as all concerned, including the engineering staffs at the Admiralty, dockyards and contractors' works, as well as the engineers of the ships, gain experience.

There is no doubt that the advance from cylindrical to water-tube boilers, with its accompanying great increase in pressures from 150 pounds to 250 pounds at the engines, has for the present added greatly to the anxieties of the engineers in charge of the machinery. This is inevitable when any change of this magnitude is made, involving, as it does, such a multitude of small details.

It should be fully recognized that these officers have at first a difficult task, and that time is necessary to enable them to gain experience in the best way of dealing with all emergencies that arise under the new conditions.

Most of the difficulties are got over after a ship has been some time in commission, especially if the service she is employed on permits of time and opportunity being given to remedy all the defects that are discovered; and the difficulties are all of such a nature that it may be confidently expected that they will be successfully overcome, by comparatively small modifications of design or manipulation, as experience is gained.

Men-of-war must be designed to cope with those of foreign countries that they may have to meet in war, and no country can afford to relinquish such a decided advantage in speed for a given weight as the trials of *Highflyer* and *Minerva* showed to be given by water-tube boilers, or the great advantage of getting up steam and increasing speed rapidly, unless there were strong grounds for supposing that the numerous defects in details which now render the machinery somewhat less reliable than older and well-tried types, were likely to be permanent.

This is certainly not the case. All the experience in our commissioned ships shows that the defects from which they at first suffered are being rapidly overcome, and it is practically certain that if we were to revert to cylindrical boilers and accept

the sacrifice of speed or increase of displacement that it would entail, we should find that before the first ship was completed all the difficulties of detail which now give us trouble in water-tube boiler ships will have been overcome, and all our new vessels would be distinctly inferior to those of all other countries, with no compensating advantages.

Taking the results of the *Highflyer's* and *Minerva's* trials as they stand, and putting the 100 tons of weight saved in the machinery of the former into horsepower instead of guns, the gain of speed due to the water-tube boilers under practical sea-going conditions cannot be put at less than a knot, a result which is borne out in a general way by the comparisons of other runs of vessels with cylindrical and Belleville boilers respectively, though in these cases equally precise conclusions cannot be drawn from them, owing to the form and dimensions of the ships compared not being identical.

Thus, if the Admiralty of the day had not decided to put water-tube boilers into the *Powerful* when she was designed, or the subsequent successive Boards had hesitated to follow their policy, and had waited until the whole of the minor difficulties involved in the change had been overcome, the magnificent fleet which has been built, building and projected since that time, consisting of—

Battleships—	6	*Canopus.*	
	8	*Formidables.*	
	6	*Duncans.*	
	20	. . . . .	20
Armored Cruisers— . . .	4	*Drakes.*	
	6	*Cressys.*	
	12	*Monmouths.*	
	22	. . . . .	22
Protected First-Class Cruisers—	2	*Powerfuls.*	
	8	*Diadems.*	
	10	. . . . .	10

Second-Class Cruisers— . .	4	*Arrogants.*
	5	*Hermes.*
	9	 9
		61

making a total of 61 ships, would have had at least a knot less speed than they now will have, or an equivalent sacrifice would have had to be made in other directions, which, to gain a knot at the high speeds now necessary, would be very considerable in amount.

TABLES IV AND V.

RESULTS OF WATER-CONSUMPTION TRIALS. MADE IN MAY AND JUNE, 1899.

Description of trial.	Percentage of full power.	Steam pressure			Pounds of water per I.H.P. per hour.			Pounds of coal per I.H.P. per hour.
		Boilers.	Engines, starboard	Engines, port.	Main engines.	Auxiliary.	All purposes.	
H. M. S. *DIANA.*								
30 hours at 800 I.H.P.	8.3	101	96	97	20.59	5.68	26.27	2.82
12 hours at 1,600 I.H.P.*a*	16.6	124	120	121	18.94	4.21	23.15	2.52
30 hours at 1,600 I.H.P.*b*	16.6	117	113	113	18.26	3.29	21.55	2.52
30 hours at 4,800 I.H.P.	50.0	133	128	128	17.09	2.14	19.23	2.02
30 hours at 6,400 I.H.P.	66.6	138	129	129	17.9	2.56	20.46	2.38
8 hours at 8,000 I.H.P.	83.3	150	144	144	17.75	1.94	19.69	2.39
H. M. S. *HERMES.*								
12 hours at 1,000 I.H.P.*c*	10.0	241	169	176	21.9	5.2	27.1	2.5
12 hours at 1,000 I.H.P.*d*	10.0	163	124	123	20.1	6.67	26.8	2.3
30 hours at 2,000 I.H.P.	20.0	196	126	127	17.06	4.22	21.28	1.76
30 hours at 7,500 I.H.P.	75.0	263	223	222	15.3	2.1	17.4	1.57
12 hours at 7,500 I.H.P.*e* (Closed exhaust system in use.)	75.0	264	228	219	...	...	16.84	1.5
8 hours at 9,000 I.H.P.*f* (Closed exhaust system in use.)	90.0	270	249	244	...	...	17.13	1.54
8 hours at 10,000 I.H.P.	100.0	240	231	228	15.64	2.05	17.69	1.58

a Cut off in H.P. cylinder 38 per cent.
b Cut off in H.P. cylinder 44.5 per cent.
c Main links linked up as far as practicable—cut off in H.P. cylinder 13.5 per cent.
d Main links in full gear—independent links run in. Cut off in H.P. cylinders 47.8 per cent.
e Auxiliary exhaust at 26 pounds pressure in use on evaporators and surplus to L.P. receivers.
f Auxiliary exhaust at 29 pounds pressure in use on L.P. receivers only.

TABLE VIII.—PASSAGES OF *DIADEM* AND *ANDROMEDA*, COMPARED WITH *BLENHEIM*, *BLAKE* AND *ROYAL ARTHUR*.

Name of vessel	Diadem.	Andromeda.	*Blenheim.*		*Blake.*	*Royal Arthur.*	U. S. S. *Columbia.*
Date	9-12, 12, 98	12-14, 4, 1900	*13-14, 6, 96*	*15-16, 6, 96*	*6-8, 5, 98*	*22-25, 3, 94*	*26, 7, 95 to 2, 8, 95*
Place	Gibraltar to the Nore.	Gibraltar to Malta.	*Stockholm to Sheerness.*		*Gibraltar to Sierra Leone.*	*Callao to Coquimbo*	*Southampton to New York.*
Total distance run, miles	1330	891.3	*466.3*	*465.6*	*1134*	*1298*	*3000*
Duration, hours	69	46 18/60	*28.5*	*26.0*	*72*	*72*	*168*
I.H.P.	14268	14264	*6980*	*8080*	*7011*	*8821*	*9500 (estimated.)*
Proportion of authorized natural draft, power, per cent.	86.4	86.4	*53.7*	*62.1*	*53.9*	*88.2*	*53*
Speed, knots	19.27	19.1	*16.3*	*17.9*	*15.7*	*18.03*	*18.4*
Revolutions per minute	110.0	107.9	*73 to 76*	*78 to 81*	*70*	*93.95*	*No record.*
Consumption of coal per I.H.P. per hour, all purposes, pounds.	2.16	2.3	*2.11*	*2.08*	*2.49*	*2.00*	*2.07 (estimated.)*
Description of coal	Welsh.	Insole's Merthyr and Dowlais Merthyr.	*Ocean Merthyr*	*Ocean Merthyr and National Merthyr.*	*West Hartley and Cyfarthfa.*	*Cambrian and Penrikyber.*	*No record.*
Quality of coal	Average.	5	*8*	*8*	*7*	*8*	*No record.*
Total coal used, tons	950	683	*187.*	*195*	*560*	*569.4*	*1474*

Names of ships with Scotch boilers, in italics.

TABLE IX.—TRIALS OF *POWERFUL* AND *TERRIBLE*.

Name of vessel.	Powerful.					Terrible.			
Reason for trial	Trial ordered at half power by Admiralty. *a*	Quarterly passage trial. *b*	Quarterly passage trial. *c*	Quarterly passage trial. *d*	4 hours' trial to replace that attempted in July, 1898. *e*	Trial at ½ power ordered by Admiralty. *f*	Trial at 18,000 I.H.P., ordered by Admiralty. *g*	Trial at 22,000 I.H.P., ordered by Admiralty. *h*	Trial at full power ordered by Admiralty. *i*
Date	27–28, 11, 97	28–29, 7, 98	7–8, 3, 98	17, 6, 99	10, 8, 98	13–16, 8, 98	29–31, 8, 98	13, 9, 98	15, 9, 98
Place	Port Louis to Colombo.	Wei-Hai-Wei to Yokohama.	Hong Kong to Manila.	Hong Kong to Wei-Hai-Wei.	Yokohama to Hong Kong.	Channel and Atlantic.	Channel and Atlantic.	English Channel	English Channel
Total distance run, miles	421.6	161.8 / 51 8 = 255.6	81 3	459.3	82.6	233.4	81.1	1176	1218	165.6	87.6
Duration, hours	24.0	8.0 / 2.5 = 13.5	4	23	4	11⅔	4	60	60	8	4
I.H.P.	12074	18589 / 21043 = 15823	21444	19298	23032	19089	23628	15554	18515	23053	25112
Proportion of authorized N.D. power, per cent.	48.3	74.3 / 84.1 = 63.2	85.7	77.2	92.1	76.3	94.5	62.2	74.1	92.2	100
Speed, knots	17.57	20.22 / 20.72 = 18.93	20.325	19.97	20.65	20.0	20.275	19.6	1st 30 hrs., 20.1 / 2d 30 hrs., 20.5	20.7	21.9
Revolutions per minute, number	90.0	102.35 / 108.55 = 97.7	105.75	102.58	108.09	104.0	110.37	93.66	98.38	104.2	108.57
Consumption of coal per I.H.P. per hour, all purposes, pounds.	2.27	1.98 / 1.99 = 1.96	2.6	2.6	2.44	2.33	2.14	1.96	2.18	2.11	2.11
Description of coal	DowlaisMerthyr Hood's Merthyr	Harris Navigation and Ocean Merthyr.	Welsh.		Insole's Merthyr		Ferndale.	Welsh.	Welsh.	Welsh.	Welsh.
Quality of coal	6 to 7	7 to 8	Small and dusty to 6.		Small and dusty to 6.		6 to 7	6 to 7	5 to 6, small and of fair quality.	6 to 7	6 to 7
Total coal used, tons	293.4	131.45 / 46.73 = 187.48	99.6	515.1	100.4	231.5	90.5	819.4	1081	174	94.6

REMARKS—DEFECTS.

a All crankheads, except P.H.P. and S.A.L.P., worked warm.
S.I.P., P.H.P., and P.I.P. indicator levers carried away.
S.F. bilge pump plunger became detached from rod.
Drain tank pump, ford piston rod broken.
Exhaust valve to atmosphere leaked badly.
S.H.P. cylinder cover, joint blowing.
Number 19 boiler, top row of fusible plugs blew out. Some joints in main and auxillary steam pipes leaking.

b Attempt to reach 25,000 I.H.P. was abandoned after 2⅓ hours, on account of inability of stokers to keep steam.
No defects.

c Could not reach 25,000 H.P. on account of small and dusty coal.
No. 9 fan, white metal run, and one bolt broken in crankhead brasses.
P. main condenser, 2 tubes leaking.

d 18,000 H.P. trial abandoned after 11⅔ hours, on account of thick fog necessitating a reduction of speed.
No. 8 fan engine white metal in crankhead brasses ran, engine stopped.

e Trial ran for 4 hours only, in accordance with instruction from C. in C.
Passage trial having been previously run, 28–29, 7, 98.
Both main condensers leaking.
Main and upper deck casings in vicinity of funnels became very hot, and paint work blistered.
25,000 H.P. not reached on account of coal being small and dusty.

f Port H.P. piston rod gland leaked slightly.
Nos. 25, 29, 30 and 45 boilers require nickel ferrules renewing.
No. 36 boiler, one element requires renewing.
No. 5 air furnace engine requires overhauling.

g P.H.P. piston rod gland leaking.
No. 29 boiler had ferrules leaking and tubes bent.
P. hot well pump suction pipe and S. hot well pump discharge pipe defective.
P.A. fire engine had plunger off rod.
Several joints in auxiliary steam pipes leaking.

h Stay bolt at bottom of H.P. cylinder (port) blowing badly.
Rings at back of both I.P. slide valves left contact with valve.

i Nickel ferrules leaked badly on Nos. 26 and 30 boilers on easing.
Lever of starboard reducing valve broken.
Trouble in adjusting rings at back of I.P. slide valves.

TABLE XI.—QUARTERLY PASSAGE TRIALS OF SHIPS WITH BELLEVILLE BOILERS.

Name of vessel.	*Canopus.*	*Andromeda.*	*Niobe.*	*Diadem.*	*Arrogant.*		*Furious.*		
Date	12–13, 4, 00 *a*	13–14, 3, 00 *b*	13–15, 3, 99 *c*	25–27, 5, 99 *d*	10–11, 9, 98 *e*	9, 12, 98 *f*	10–11, 9, 98 *g*	9–10, 12, 98 *h*	9–10, 2, 00 *i*
Place	Spezzia to Malta	Malta to Naples	English Channel	Arosa Bay to Sheerness.	Bantry to Plymouth.	Gibraltar to Plymouth.	Bantry to Portland.	Gibraltar to the Downs.	Las Palmas to England.
Total distance run, miles	375.7	77.4 358.6	1050	777	82.8 355.8	76.3	81.5 355.5	75.8 325.6	354.5
Duration, hours	24	4 20	60	42	4 20	4	4 20	4 20	24
I.H.P.	8163	14,213 9,993	10,031	13,517	10,052 6,275	9052	10,359 6,292	10,405 6,260	5988
Proportion authorized natural-draft power, per ct.	60.4	86.1 60.0	60.5	81.8	100.5 62.7	90.5	103.6 62 9	104.0 62.6	59.9
Speed, knots	15.6	19 35 17.93	17.5	18.5	20.7 17.79	19.07	20.375 17.775	18.95 16.28	14.77
Revolutions per minute, number.	93.1	113.5 100	98.55	106.2	141.2 123.25	140	140.59 122.4	139.66 120.93	115.1
Coal per I.H.P. per hour, all purposes, pounds.	2.36	2.10 2.18	2.046	2.33	2.67 2.36	3.3	2.91 2.49	2.6 2.32	2.79
Description of coal	Hood's Merthyr, Locket's Merthyr, Penrikyber Navigation.	Cyfarthfa Merthyr.	Standard Merthyr and Cyfarthfa Merthyr.	Cambrian Navigation and Dowlais Cardiff.	Cambrian and Lewis Merthyr.	Wayne's Merthyr, Plymouth Merthyr, Cory's Merthyr.	Abercorn black vein.	Powell Duffryn.	Powell Duffryn.
Quality of coal	Very small, 6 to 8	7	6 to 7	Average.	7 to 8	3 to 4	7	8	6 to 7
Total coal used, tons	206.7	53.4 195	550	591	48 132	53.6	54 140	48.4 130	178.7

REMARKS—DEFECTS.

a Both H.P. piston-rod glands blowing.
Several joints in main and auxiliary steam pipes leaking.

b White metal of P.I.P. crankhead run.

c Nos. 14 and 26 boilers, steady pipe connections leaky.
Fusible plugs in No. 28 boiler required renewal, tubes bent.
S.H.P. crankhead and No. 4 main bearing P. worked warm.
S.H.P. piston-rod gland leaked badly.

d S.I.P. crank and crosshead brasses warmed up.
P. main air pump, guide bar cracked. Three nickel ferrules renewed.

e H.P. piston-rod glands leaked.
Several fusible plugs required renewal.
Auxiliary air-pump discharge pipes defective.

f Tube burst in No. 8 boiler.
Many leaky nipples in No. 15 boiler.
H.P. piston-rod glands leaked badly.
S.L.P. crankhead worked warm.
Feed pumps worked very badly and needed manual assistance.

g Joints of L.P. cylinder covers leaky.

h One fan engine broke down.

i No defects,

TABLE XIII.—PARTICULARS OF TUBES RENEWED IN BELLEVILLE BOILERS.

Name of ship.	Total number of tubes fitted.		Generators.	Economizers.	Down-cast.
	Generator.	Economizer.			
Sharpshooter..	1,080	Nil	6 tubes (defective.)	.....	Nil
Hermes.....	2,268	1,944	A number of tubes defective and now being renewed. The following have been ordered and sent to Bermuda: 54 elements 460 tubes	18 elements	.....
Terrible.....	7,200	Nil	7 elements 78 tubes	.....	5 tubes
Diadem.....	3,332	2,576	8 tubes (defective.)	5 tubes (defective.)	Nil
Furious.....	2,880	Nil	6 tubes (defective.)	.....	Nil
Europa.....	3,332	2,576	1 tube (defective.)	Nil	Nil
Arrogant.....	2,880	Nil	35 tubes (defective.)	.....	Nil
Powerful.....	7,200	Nil	11 elements 35 tubes	.....	20 tubes
Niobe.....	3,332	2,576	1 tube (defective.)	Nil	Nil
Andromeda...	3,332	2,576	Nil	Nil	Nil
Highflyer.....	2,268	1,944	Nil	Nil	Nil
Vindictive.....	2,880	Nil	Nil	Nil	Nil
Gladiator.....	2,880	Nil	Nil	Nil	Nil
Argonaut.....	3,192	4,560	Nil	Nil	Nil
Goliath.....	2,450	2,600	Nil	Nil	Nil
Canopus.....	2,450	2,600	Nil	Nil	Nil
Ocean.....	2,450	2,600	Nil	Nil	Nil

NOTE.—An element consists of 14 or 20 tubes in the generators and from 12 to 20 in economizers, but in many cases a whole element is removed for convenience when only one or two tubes are defective.

TABLE XIV.—TOTAL NUMBER OF BELLEVILLE-BOILER TUBES (EXCLUDING *HERMES*) WHICH HAVE FAILED SINCE THE ACCIDENT IN THE *TERRIBLE* IN MARCH, 1899.

Name of ship.	Generator tubes.	Economizer tubes.	Downcast tubes.	Total.
Powerful.....	7	Nil	10	17
Terrible.....	Nil	Nil	Nil	Nil
Diadem.....	8	5	Nil	13
Furious.....	3	Nil	Nil	3
Arrogant.....	4	Nil	Nil	4
Europa.....	1	Nil	Nil	1
Total.....	23	5	10	38

Excluding *Hermes*, there have been only 38 tubes renewed in the whole fleet since the accident in *Terrible* of March, 1899. Of these, 10 were downcomer tubes; their decay is attributed to defective design; 13 were the pitted tubes in *Diadem*; and 9 were due to defective material, namely, 5 lap-welded tubes and 4 solid-drawn of the earliest manufacture.

Appendix No. 1.

Defects of Highflyer.—

Ship docked, stern shafts, bearings, propellers and fastenings examined.

Two main bulkhead valves and six sectional valves refitted.

Joints of main steam pipes and stop-valve covers in each engine room remade.

Refitted both H.P. piston rod metallic packing.

Two chocks under engines renewed.

Water-service pipes repaired.

Eight guide slippers remetalled and refitted in place.

Six end doors of steam collectors rejointed.

Smoke-box doors refitted and supports stiffened.

Leak in steam collector No. 12 boiler made good.

Ash-pit water-service cocks refitted with two plugs.

Pressure gages tested.

Several water pipes repaired and partly renewed.

Sluice valves on submerged torpedo tubes overhauled and refitted.

One new stay with plus threads at ends fitted in cylinder cover. New guard rings in one piece fitted to secure the junk-ring nuts (in lieu of the existing one which was in segments), and secured by square-necked studs, and nuts and split pins.

Valves in connection with the steam-drain service made as far as possible separate from the spindles. Asbestos-packed cocks in connection with the exhaust-drain service refitted. Some pipe flanges refaced, and thicker fitted where considered necessary, and rejointed with red lead and fitted bolts.

Four new working and one spare L.P. piston rings, with tongue pieces complete, fitted.

Faces of L.P. slide valves faced, and new hard cast-iron faces secured to cylinders with new gun-metal screws.

Main air pumps overhauled and refitted with new guards and head valves as necessary.

Several auxiliary stop valves fitted with new valves and spindles.

Distilling pumps overhauled and defects made good.

Channel-steel pillars supporting back shelf under generator elements of boilers renewed.

Copper-wire gauze supplied for remaking joints.

Alterations and Additions made in Highflyer.—

Fitted two additional evaporators with feed and brine pumps.

Additional liners fitted to main engine astern guides.

Check nuts fitted to bolts of link gear.

Worsted holders and wipers fitted to crank and crosshead-bearing lubricating arrangements.

Drains from fire-engine steam pipes led to trap in auxiliary service.

Casings fitted over valve spindles in stokehold bilges.

Means provided for taking weight of elements when being removed or replaced.

Additional lubricator boxes fitted to crank pins on opposite side of engines to existing lubricators.

Position of valves for tube-sweeping apparatus altered.

Relief valves fitted on suction air vessels of the feed-suction pipes.

All joints on main steam pipes in engine room remade with red-lead paint.

Boiler-door joints remade with continuous asbestos rings.

New steam stop-valves boxes with thicker flanges fitted to the feed, fire and hot-well engines.

Stop pins fitted to the stay studs of expansion glands in auxiliary steam pipes.

Guides of automatic feed-float spindles bushed with gun-metal.

Six spare wrought-steel connections for water-gage standards, and feed boxes to junction boxes made and supplied.

Stops fitted to air-pump bucket and crosshead ends of rod, also to air-pump guide bushes.

Stay fitted to forward end of main run of auxiliary steam pipe in stokehold.

Four new distiller pumps fitted in place.

Traps removed from evaporators, and coil drains led to auxiliary condensers. The same traps refitted and fitted in boiler rooms to drains from sectional valves on boilers.

All drains removed from stop valves of center boilers in each stokehold.

Drains from steam pipes to feed engines removed from auxiliary drain service and fitted to main.

Safety-valve drains removed from drain tank and fitted to separate tank in stokehold.

Drains from fans and blowers removed and blanked.

Drains from dynamos and air compressors removed and blanked.

Vapor pipe from drain tank led into air shoot and to the auxiliary condensers with stop valves as necessary in the cross connection.

Removed drains from bulkhead stop valves, intermediate valves, and middle-line-bulkhead valves in each engine room and trapped same in engine rooms.

Joints of auxiliary steam pipes to refrigerator and capstan engine in refrigerator flat faced and remade with red-lead paint.

Several joints in main and auxiliary steam pipes in stokehold refaced and rejointed with red-lead paint.

APPENDIX No. 2.

TYPES OF WATER-TUBE BOILERS FITTED AND TO BE FITTED IN WAR VESSELS OF THE PRINCIPAL FOREIGN NAVIES.

N. B.—In the case of vessels stated to have been re-boilered or to be re-boilered, the water-tube boiler has been, or will be, substituted for the cylindrical boiler.

Name.	Date of launch.	I.H.P.	Type of boiler.
FRANCE.			
BATTLESHIPS—BUILT.			
Gaulois...	1896	14,500	Belleville.
Bouvet	1896	14,000	Belleville.
Charlemagne	1893	14,500	Belleville.
Masséna	1893	15,000	Lagrafel d'Allest.
Carnot	1894	15,000	Lagrafel d'Allest.
Charles Martel	1893	13,600	Lagrafel d'Allest.
Jauréguiberry	1893	15,000	Lagrafel d'Allest.
Amiral Tréhouart	1893	7,500	Belleville.
Bouvines	1892	9,000	Lagrafel d'Allest.
Valmy	1892	9,000	Lagrafel d'Allest.
Jemmapes	1892	9,000	Lagrafel d'Allest.
Brennus	1891	14,000	Belleville.
Neptune	1887	11,090	Belleville, to be re-boilered.
Marceau	1887	12,000	Niclausse, to be re-boilered, 1900–1901.
Hoche	1886	11,836	Belleville, re-boilered, 1899.
Requin	1885	7,000	Niclausse, re-boilered, 1899.
Amiral Duperré	1879	11,032	To be re-boilered; type not yet known.
Dévastation	1879	8,301	Belleville, to be re-boilered, 1900–1901.
BATTLESHIPS—BUILDING.			
St. Louis	1896	14,500	Belleville.
Jéna	1898	15,500	Belleville.
Suffren	1899	16,200	Niclausse.
Henri IV	1899	11,500	Niclausse.
CRUISERS, ARMORED—BUILT.			
Bruix	1894	8,800	Belleville.
Chanzy	1894	7,400	Belleville.
Amiral Charner	1893	7,400	Belleville.
Latouche-Tréville	1892	7,400	Belleville.

Name.	Date of launch.	I.H.P.	Type of boiler.
CRUISERS, PROTECTED—BUILT.			
Infernet	1899	8,500	Normand.
Protet	1898	9,000	Belleville.
Chateaurenault	1898	23,000	Normand-Sigaudy.
Guichen	1897	24,000	Lagrafel d'Allest.
Lavoisier	1897	6,600	Belleville.
D'Estrées	1897	8,500	Normand.
Catinat	1896	9,000	Belleville.
Fleurus	1893	4,000	Niclausse.
Galilée	1896	6,600	Belleville.
Cassard	1896	8,500	Lagrafel d'Allest.
D'Assas	1896	8,500	Lagrafel d'Allest.
DuChayla	1895	8,500	Lagrafel d'Allest.
Pothuau	1895	10,000	Belleville.
Pascal	1895	8,500	Belleville.
Descartes	1894	8,500	Belleville.
Bugeaud	1893	9,000	Belleville.
Chasseloup-Laubat	1893	8,000	Lagrafel d'Allest.
Friant	1893	9,000	Niclausse.
Suchet	1893	9,000	Belleville, re-boilered, 1898.
Alger	1889	8,000	Belleville, f'm Lagrafel d'Allest
Davout	1889	3,031	Niclausse; being reboilered.
Tage	1886	7,469	To be re-boilered; type not known.
CRUISER, UNPROTECTED—BUILT.			
Milan	1884	3,800	Belleville.
CRUISERS ARMORED—BUILDING.			
Jeanne d'Arc	1899	28,500	Guyot.
Gueydon	1899	19,600	Niclausse.
Dupetit Thouars	1899	19,600	Belleville.
Montcalm	1900	19,600	Normand-Sigaudy.
Dupleix	1900	17,100	Belleville.
Desaix	...	17,100	Belleville.
Kléber	...	17,100	Niclausse.
Condé	...	20,500	Niclausee.
Gloire	1900	20,500	Niclausee.
Sully	...	20,500	Belleville.
Marseillaise	...	20,500	Belleville.
Amiral Aube	...	20,500	Belleville.
Leon Gambetta	...	24,000	Type not known.
Jules Ferry	...	24,000	Type not known.
CRUISERS, PROTECTED—BUILDING.			
Jurien de la Gravière	1899	17,400	Guyot.

Name.	Date of launch.	I.H.P.	Type of boiler.
COAST DEFENCE VESSELS, ARMORED.			
Achéron	1885	1,600	To be re-boilered; type not known.
Fusée	1884	1,500	Type not known; re-boilered, 1899–1900.
Furieux	1883	5,000	To be re-boilered, 1900. Belleville.
Tempête	1876	3,600	To be re-boilered; type not known.
Tonnerre	1775	3,600	Type not known; re-boilered 1899–1900.
TORPEDO TRANSPORT.			
Foudre	1895	11,000	Lagrafel d'Allest.
RUSSIA.			
BATTLESHIPS–BUILT.			
Emperor Nicolas I	1889	8,000	Belleville; reboilered, 1899.
Emperor Alexander II	1887	...	Belleville; proposed.
Sinop	1887	13,000	Belleville; to be fitted.
Ekaterina II	1886	9,000	Belleville; to be fitted.
BATTLESHIPS—BUILDING.			
Peresvyet	1898	14,500	Belleville.
Oslyabya	1898	14,500	Belleville.
Pobyeda	1900	14,500	Belleville.
Prince Potemkin	.....	10,600	Belleville.
Retvizan	.....	17,000	Niclausse.
Borodino	.....	16,300	Belleville.
Tsesarevitch	.....	16,300	Belleville.
Emperor Alexander III	.....	16,300	Belleville.
Orel	.....	16,300	Belleville.
CRUISERS, ARMORED—BUILT.			
Rossiya	1896	17,000	Belleville.
Pamyat Azova	1888	.....	Belleville (under consideration to be re-boilered).
Minin	1878	5,290	Belleville.
CRUISER, PROTECTED—BUILT.			
Svyetlana	1896	8,500	Belleville.
CRUISERS, ARMORED—BUILDING.			
Gromoboi	1899	14,500	Belleville.
Bayan	.....	16,500	Belleville.

Name.	Date of launch.	I.H.P.	Type of boiler.
CRUISERS, PROTECTED—BUILDING.			
Diana	1899	11,610	Belleville.
Pallada	1899	11,610	Belleville.
Varyag	1899	20,000	Niclausse.
Askold	1900	19,000	Schültz.
Aurora	1900	11,610	Belleville.
Bogatyr	.....	19,500	Normand.
Novik	.....	17,000	Schichau.
Boyarin	.....	11,500	Belleville.
GERMANY.			
BATTLESHIPS—BUILT.			
Kaiser Wilhelm II.	1897	13,000	⅓ Schülz; ⅔ cylindrical.
Kaiser Friedrich III.	1896	13,000	⅓ Thorncroft; ⅔ cylindrical.
Aegir	1895	5,000	Thornycroft.
Hagen	1893	5,000	Thornycroft, re-boilered, 1899.
Baden	1880	6,000	Dürr, re-boilered, 1897.
Bayern	1878	6,000	Dürr, re-boilered, 1898.
Württemberg	1878	7,000	Schülz, re-boilered, 1899.
Sachsen	1877	6,000	Dürr, re-boilered, 1899.
BATTLESHIPS—BUILDING.			
Kaiser Wilhelm der Grosse.	1897	13,000	½ Schülz; ½ cylindrical.
Kaiser Karl der Grosse	1899	13,000	½ Schülz; ½ cylindrical.
Kaiser Barbarossa	1900	13,000	½ Thornycroft; ½ cylindrical.
Wittelsbach	1909	15,000	½ Schülz; ½ cylindrical.
D	.....	15,000	½ Thornycroft; ½ cylindrical.
E	.....	15,000	½ Schülz; ½ cylindrical.
F	.....	15,000	½ Schülz; ½ cylindrical.
G	.....	15,000	½ Schülz; ½ cylindrical.
CRUISER, ARMORED—BUILT.			
Fürst Bismarck	1897	15,000	⅓ Schultz ⅔ cylindrical.
CRUISERS, PROTECTED—BUILT.			
Nymphe	1899	8,000	Schülz.
Niobe	1899	8,000	Thornycroft.
Gazelle	1898	6,000	Niclausse.
Hansa	1898	10,000	Belleville.
Vineta	1897	10,000	Dürr.
Freya	1897	10,000	Niclausse.
Victoria Louise	1897	10,000	Dürr.
Hertha	1897	10,000	Belleville.
CRUISER, ARMORED—BUILDING.			
Prinz Heinrich	1900	15,000	Dürr.

Name.	Date of launch.	I.H.P.	Type of boiler.
CRUISERS, PROTECTED—BUILDING.			
Thetis	1900	8,000	Schültz.
D	.....	8,000	Schültz.
E	.....	8,000	Schültz.
F	.....	8,000	Schültz.
ITALY.			
BATTLESHIPS—BUILDING.			
Benedetto Brin	.....	19,000	Belleville.
Regina Margherita	.....	19,000	Niclausse.
CRUISERS, PROTECTED—BUILT.			
Agordat	1899	7,000	Blechynden.
Coatit	1899	7,000	Blechynden.
CRUISERS, ARMORED—BUILDING.			
G. Garibaldi	1899	13,500	Niclausse.
Varese	1899	13,500	Belleville.
F. Ferruccio	.....	13,500	Belleville.
CRUISERS, ARMORED—PROJECTED.			
1 (type *Bettolo*)	.....	19,000	Type not known.
1 (type *Bettolo*)	.....	19,000	Type not known.
JAPAN.			
BATTLESHIP—BUILT.			
Shikishima	1898	14,500	Belleville.
BATTLESHIPS—BUILDING.			
Asahi	1899	15,000	Belleville.
Hatsuse	1899	14,500	Belleville.
Mikasa	.....	15,000	Belleville.
CRUISERS, ARMORED—BUILT.			
Chiyoda	1899	5,700	Belleville, re-boilered, 1899.
Yakumo	1899	15,500	Belleville.
CRUISER, PROTECTED—BUILT.			
Itsukushima	1889	.....	Belleville, to be re-boilered.
CRUISERS, ARMORED—BUILDING.			
Adzuma	1899	17,000	Belleville.
Idzumo	1899	14,500	Belleville.
Iwate	1900	14,500	Belleville.
CRUISER, PROTECTED—BUILDING.			
Chihaya	1900	6,000	Thornycroft.

Name.	Date of launch.	I.H.P.	Type of boiler.
HOLLAND.			
CRUISERS, PROTECTED—BUILT.			
Utrecht	1898	10,000	Yarrow.
Gelderland	1897	10,000	Yarrow.
Noord Brabant	1897	10,000	Yarrow.
Zeeland	1897	10,000	Yarrow.
Holland	1896	10,000	Yarrow.
Friesland	1896	10,000	Yarrow.
CRUISERS, ARMORED—BUILDING.			
Koningin Regents	.....	6,000	Yarrow.
De Ruyter	.....	6,000	Yarrow.
AUSTRIA.			
CRUISER, ARMORED—BUILT.			
Kaiser Karl VI	1898	12,900	Belleville.
CRUISER, PROTECTED—BUILT.			
Zenta	1897	7,200	Yarrow.
CRUISER, ARMORED—BUILDING.			
E	.....	13,000	Yarrow.
CRUISERS, PROTECTED—BUILDING.			
Aspern	1899	7,200	Yarrow.
C	.....	7,200	Yarrow.
COAST-DEFENCE VESSEL, ARMORED—BUILT.			
Budapest	1896	9,185	Belleville.
COAST-DEFENCE VESSELS, ARMORED—BUILDING.			
I	.....	11,900	Belleville.
II	.....	11,900	Belleville.
SWEDEN.			
COAST-DEFENCE VESSELS, ARMORED—BUILT.			
Thor	1898	3,650	Yarrow.
Niordh	1898	3,650	Yarrow.
COAST-DEFENCE VESSELS, ARMORED—BUILDING.			
Dristigheten	1900	5,400	Yarrow.
A	.....	5,500	Yarrow.
B	.....	5,500	Yarrow.
C	.....	5,500	Yarrow.
NORWAY.			
COAST-DEFENCE VESSELS, ARMORED—BUILDING.			
Norge	1900	4,500	Yarrow.
Eidsvold	1900	4,500	Yarrow.

EFFICIENCY OF THE RECEIVER SUPERHEATER.

Mr. L. Lenke has contributed an interesting paper to the "Engineer" of London, which will be advantageously read in connection with Dr. Thurston's article on "Reheaters" published in the last issue of the JOURNAL. It is as follows:

The advantages in economy when using superheated steam is to-day an undoubted fact, and apparent to anyone who has an opportunity of investigating the matter.

Saturated steam at a boiler pressure of 140 pounds has a temperature of 361 degrees Fahrenheit; 1 pound of this steam contains a total heat of 1,192 B.T.U., and its volume is 2.92 cubic feet. By the addition of 153 B.T.U. the temperature of the steam will be increased to 610 degrees Fahrenheit if the pressure remains the same. The effect is, if the steam can expand, to increase its volume to 4.06 cubic feet, or to increase the pressure to about 210 pounds per square inch if the volume of the steam remains the same. It is impossible with boiler plants to allow the pressure to rise above a fixed point, and therefore the volume must increase. Consequently, it will be seen that when supplying 12.8 per cent. more heat—153 B.T.U. compared with 1,192 B.T.U.—the volume will be increased 39 per cent. A certain gain must obviously be obtained by superheating the steam, and the higher the superheat the greater the gain, but the actual gain in practice when using superheated steam is not only the increasing of the volume of the steam, but also the complete removal of loss by cylinder condensation. When saturated steam is used no transmission of heat can take place from the steam to the cylinder walls without condensing the steam, and it is known that a part of this condensed steam reevaporates during the exhaust period, taking the necessary amount of heat from the walls, and so cooling them and making the mean temperature of same less than that of the admission steam. On the other hand, when the

steam is sufficiently superheated, the amount of heat transferred to the cylinder walls will not be able to condense the steam, as the peculiarity of superheated steam enables it to lose a certain amount of heat without condensing, and it goes without saying that the mean temperature of the cylinder walls is higher when using superheated steam. As a matter of fact, from trials made by Professor Doerfel ("Zeitschrift des Verenes Deutscher Ingenieure," 1899, page 1065) the losses by condensation of the steam during the admission period depend more upon the moisture in the steam than the temperature, and therefore this condensation cannot be materially decreased, as some people are inclined to believe, by high compression.

By using superheated steam in engines with more than one cylinder, the condensation of the steam is almost done away with in the high-pressure cylinder, as a superheat of about two hundred degrees Fahrenheit above the boiling point is required to obtain dry steam at the beginning of expansion. The low-pressure cylinder is provided with saturated steam, a considerable portion of which condenses again. To minimize this loss Mr. Schmidt invented the receiver superheater, and this reduces the condensation in the low-pressure cylinder to almost nil. To show the importance of this fact I refer to later trials made by Professor Doerfel, but before considering them it is necessary to explain the principles of the calculations introduced by him. To enable a general comparison to be made of the steam consumption resulting with the various types of engines working with superheated and saturated steam, it is desirable to add to the usually given figures of pounds of steam per indicated horsepower per hour the calculations of the heat in B.T.U. required. Professor Doerfel proposes therefore to give the total heat, including the superheat expressed by the formula:

$$H = 1082 + 0.305t + 0.48\,(t_2 - t_1).$$

A deduction for the feed-water temperature and the heat in the exhaust pipe is not to be recommended, as these figures vary according to the special circumstances of each case, and the first one particularly is influenced to a large degree when heaters are

used. The following figures in the subjoined table of results are calculated on this principle. The advantages to be expected from superheating the receiver steam before it reaches the low-pressure cylinder are very clearly shown from the trials given with different temperatures at a very low steam pressure.

The engine on which Professor Doerfel experimented is a Corliss condensing single-cylinder engine, with a cylinder 18 inches in diameter, stroke 35½ inches, running at seventy-four revolutions per minute.

It will be seen from these figures that the steam consumption per indicated horsepower in B.T.U. is less the more the superheat, but it may be noted that the higher the degree of superheat the smaller is the percentage of gain.

The amount of heat in the steam in trial No. 3 above that in trial No. 1 is 26.66 B.T.U., and this effects a saving of 58.5 B.T.U. per indicated horsepower. Therefore 1 B.T.U. of heat in the steam saves 2.2 B.T.U. per indicated horsepower. Now compare No. 1 with No. 5. Then 53 B.T.U. of heat saves 102.5 B.T.U. per indicated horsepower, and therefore 1 B.T.U. saves 1.93 B.T.U. per indicated horsepower. No. 1 in comparison with 10 shows that 108.2 B.T.U. more heat in the steam save 163 B.T.U. per indicated horsepower, therefore 1 B.T.U. of heat saves 1.51 B.T.U. per indicated horsepower. Compare the first and last results, and it is seen that 1 B.T.U. saves 2.2 B.T.U. per indicated horsepower as against 1.51 B.T.U., that is to say, 0.69 B.T.U. less = 31 per cent.

The foregoing figures show that the saving effected when the whole of the superheat is passed through the high-pressure cylinder cannot be so great as when dividing the amount of heat into two parts, one part—which should only be sufficient to avoid condensation—going to the high-pressure cylinder, and the remainder of the superheat to the low-pressure cylinder for the same purpose.

Two further advantages are derived from this arrangement. First, we are able to work with highly superheated steam without running the risk of overheating the high-pressure cylinder, as the steam cooled in the receiver to such a temperature as is suitable

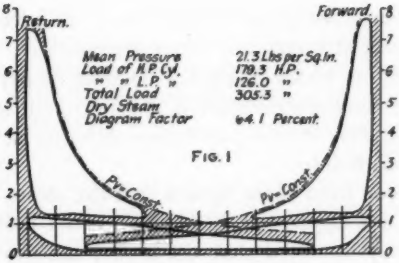


FIG. 1

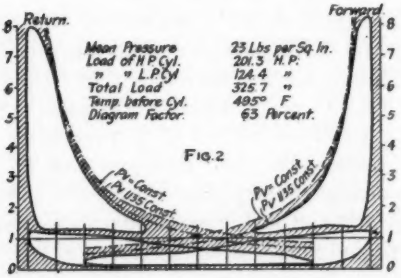


FIG. 2

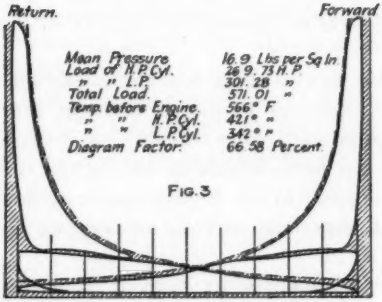

Return.
Forward
Mean Pressure 16.9 Lbs per Sq In
Load of H.P.Cyl. 269.75 H.P.
" " L.P. 301.28 "
Total Load 571.01 "
Temp. before Engine. 566° F
" " H.P.Cyl. 421° "
" " L.P.Cyl. 342° "
Diagram Factor: 66.58 Percent.

FIG. 3

for working in the cylinder, and this is especially so if the amount of heat is regulated according to the various loads—cut-offs in the high-pressure cylinder—of the engine. Secondly, as the low-pressure steam is superheated, it effects an increase in the work of the low-pressure cylinder either by increasing the pressure in the receiver or allowing a later cut-off.

It is known from various tests that for a normal cut-off in the high-pressure cylinder about 500 degrees Fahrenheit steam temperature is required to avoid any condensation, but as superheated steam with this temperature has a volume about 23 per cent. above saturated steam of the same pressure the volume of the steam entering the low-pressure cylinder is about 23 per cent. less, as it is only dry, and roughly, therefore, the low-pressure cylinder must do about 23 per cent. less work. To enable the comparison to be understood, reference should be made to the three Rankinized diagrams given showing the influence of superheat applied to the high-pressure cylinder only, and the effect of superheat with the receiver superheater.

The first diagram, No. 1, is taken from an engine supplied with dry or slightly superheated steam; No. 2 is taken from the same engine supplied with superheated steam of about 495 degrees Fahrenheit, but without superheater receiver; and the third diagram is taken from another engine in which the admission steam passes through the receiver before entering the high-pressure cylinder, so as to superheat the receiver steam. In the first diagram the proportion of work done in the two cylinders is 179 divided by $126 = 1.42$. In diagram No. 2 the proportion is 201 divided by $124 = 1.62$.

It will be seen that when the temperature is increased the high-pressure cylinder does more work proportionately than the lower-pressure cylinder. The distribution of steam which may be seen from these diagrams shows the peculiar features of superheated steam compared with saturated steam. The drop at the end of expansion compared with the hyperbolic line is greater with superheated steam than with saturated steam. Hence the difference in the work done in each cylinder, as above referred to.

Diagram No. 3 is taken from an engine having a high-pressure

RESULTS OF PROFESSOR DOERFEL'S EXPERIMENTS.

	1.	2.	3.	4.	5.	6.	7.	8.	9.	10.
Temperature of superheated steam before entering the cylinder, degrees, Fahrenheit	277	316	366	379	392	412	432	446	473	509
Boiling point, degrees, Fahrenheit	230.9	230	241.7	249.8	244.4	235.4	241.3	242.6	248	251.5
Steam pressure, absolute, pounds	21.58	21.28	26.1	30.12	27.6	23.6	26.06	26.85	29.27	31.68
Degrees of superheat, Fahrenheit	46.1	.86	94.3	129.2	147.6	176.6	190.6	203.4	225	257.5
Consumption of feed water per I.H.P. hours, pounds	29.86	28.05	26.27	24.11	23.59	23.12	22.35	22.44	20.52	19.72
Total heat of feed-water according to temperature and pressure, B.T.U.	1,174.3	1,193.6	1,200.9	1,220.2	1,227.4	1,238.4	1,246.8	1,253.7	1,265.5	1,282.5
Consumption in B.T.U. per I.H.P. minute	584.5	558	526	490	482	477	464	468	433	421
Indicated horsepower	29.79	29.03	37.58	44.88	41.87	34.30	37.65	38.90	46.45	49.57
Additional heat above trial No. 1 in B.T.U	.....	19.3	26.6	45.9	53.1	64.1	72.5	79.4	91.2	108.2
Saving compared with No. 1 in B.T.U.	.....	26.5	58.5	94.5	102.5	107.5	120.5	116.5	151.5	163.5
1 B.T.U. of heat saves in steam, B.T.U.	.....	1.38	2.2	2.06	1.93	1.67	1.66	1.46	1.66	1.51

cylinder 28½ inches diameter, and a low-pressure cylinder 47½ inches diameter, with a 51-inch stroke, and running at a speed of 74 revolutions per minute. The consumption of feed water was 11.6 pounds per indicated horsepower, and the heat consumption 256 B.T.U. per indicated horsepower per minute. The temperature of the steam before entering the receiver was 540 degrees Fahrenheit, and after passing the receiver, before entering the high-pressure cylinder, 405 degrees Fahrenheit. The receiver steam was superheated above its boiling point, 219 degrees Fahrenheit, to 327 degrees Fahrenheit. The steam pressure was 100 pounds per square inch, and the cut-off 10 per cent. of the stroke.

The diagram shows the weight of steam at the various points in proportion to the total capacity of the engine, as follows: 8.6 pounds per indicated horsepower per hour at the beginning of the expansion, and 9.54 pounds per indicated horsepower per hour at the end of expansion in the high-pressure cylinder, and 11.3 pounds at the beginning of expansion in the low-pressure cylinder. A superheat of 108 degrees Fahrenheit, with the low pressure of the steam in the receiver, and the large dimensions of the engine, were sufficient to prevent condensation in the low-pressure cylinder during the admission period.

It is worth mentioning that the loss in the temperature of the admission steam in the receiver is equal to the rise in the temperature of the receiver steam, and it can therefore be taken for granted that the steam is almost dry when it leaves the high-pressure cylinder. In the diagram the expansion lines very nearly reach the hyperbolic lines. The expansion lines of the high-pressure cylinder almost reach the Mariotte, which shows that the temperature in the high-pressure cylinder was too low.

That in practice the gain above referred to derived from the superheater receiver is remarkable will be seen from the two following trials:

A Sulzer triple-expansion engine in the Central Electric Station in Berlin, which is connected with a direct-fired superheater —Schmidt system—shows a steam consumption of 9.4 pounds of steam per indicated horsepower per hour, equal to 205 B.T.U. per indicated horsepower per minute. The temperature of the

steam was 613 degrees Fahrenheit in the high-pressure cylinder at a pressure of 192 pounds, and with a load of 2,907.8 indicated horsepower. Another engine at Pabianice, Russian Poland, on the Schmidt system, a double-acting twin tandem engine with superheater receiver, working with 132 pounds per square inch steam pressure and a temperature of 680 degrees Fahrenheit before entering the receiver, with a load of 1,046 indicated horsepower, gave a steam consumption of 8.95 pounds per indicated horsepower, equal to 197 B.T.U.

The tabular statement already referred to shows that very little superheat is necessary with a small pressure in the low-pressure cylinder of compound engines to effect a considerable saving. A superheat of 40.5 degrees Fahrenheit is required to obtain an economy of 6.1 per cent. in steam, and by superheating to 147 degrees Fahrenheit above saturated steam of the same pressure, a saving of 17.4 per cent. results.

Some very important trials, showing the advantages of the receiver superheater, were made in Alsace, and the results proved that under equal conditions a compound engine with receiver superheater is more economical than a triple-expansion engine with superheated steam in the high-pressure cylinder only. At these trials a triple-expansion engine was first tested with superheated steam in the high-pressure cylinder, and afterward the intermediate cylinder was removed, and a superheater receiver used in its place. With the triple-expansion engine the steam consumption was 11.3 pounds steam per indicated horsepower, as against 10.1 pounds with the same engine compounded. The total load of this engine in the first case was 746 indicated horsepower, and in the latter case 727 indicated horsepower, with a steam pressure of about 160 pounds in both trials. The saving in coal was 6 per cent. more with the compound engine, as compared with the triple-expansion engine.

RADIAL VALVE GEARS.

One of the most recent valve gears of the radial type which have been successfully applied to marine engines is that invented by Lincoln A. Lang, of Yule, N. Dak.

As a matter of record, as well as general interest, we give an illustration of the arrangement of the gear, together with a description of its operation. A photograph is also presented showing the manner in which the gear was applied to the engines of the *Zenith City*.

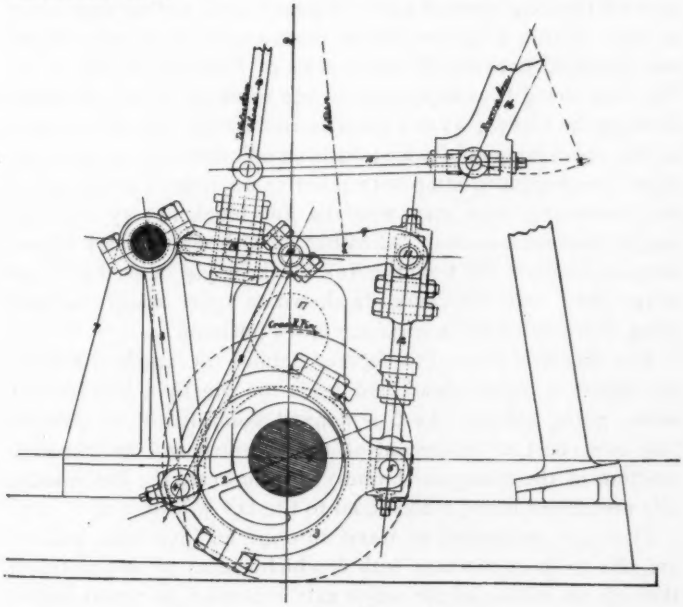


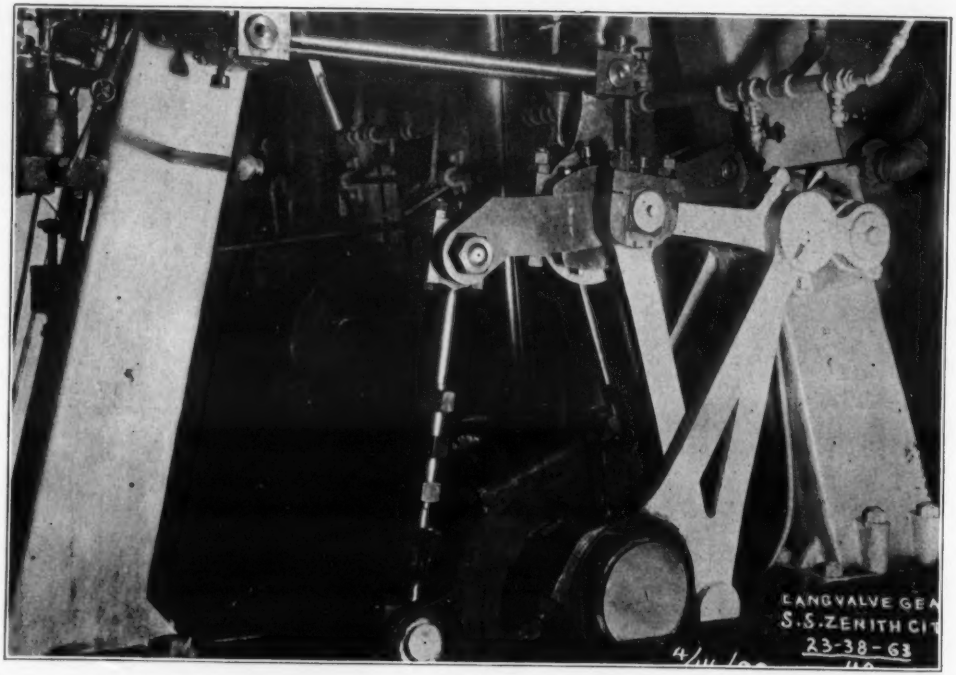
FIG. 1.

In the accompanying tracing, Fig. 1 shows an end elevation of the gear as applied to marine engines. In this view 1 is the

main shaft; 2, an eccentric keyed thereon; 3, the eccentric strap, carrying eyes 4 and 5 as shown; 6, a fixed shaft which in compound engines may, if desired, extend the length of the bedplate and parallel with the main shaft. It is supported and maintained rigidly in place by means of stools bolted to the bedplate at intervals (one of these being shown at 7), or by means of brackets cast with the main columns. Upon the shaft 6 (which constitutes the main fulcrum pin of the gear) are mounted a pair of bell cranks or triangular frames, one of which is seen at 8; the other, being behind, does not appear. These bell cranks, or frames, may be cast from steel in one piece if desired, a pair being requisite for each cylinder, all of them being mounted on the shaft 6 as shown. The eye 4 on eccentric strap is pivoted to and between the long arms of a pair of these frames so that they move as one. A link 9 carries central trunnions on each side, which are journaled between the short arms of frames as shown at 10. The link being thus supported is free to swing on its trunnions between the frames; 11 is a pivot carried by the link and located on the mean tangent of same; 12 is an adjustable rod connecting pivot 5 on eccentric strap with pivot 11 on link; 13 is a radius bar connecting valve stem with the link block 14; 15 is a drag link or bridle connecting radius bar 13 with reverse arm 16 and adapted to place and hold the link block in any desired position in the link; 17 is the crank pin shown on upper center, the gear being represented in its correct relative position.

The eccentric is set 112 degrees behind the crank pin when the engine is going ahead, and leads the pin by a like amount when going astern. As 6 is a fixed point, it will be obvious that one effect of movement of the eccentric will be (through medium of the triangular frames) to reciprocate the link bodily, this movement being made equal to the lap + lead × 2.

Pivot 4 is restrained to travel through an invariable path or arc, due to its connection with 8, which causes pivot 5 to travel through an ellipse whose major axis is parallel to center line of engine a. This movement being transmitted to pivot 11 by the rod 12 causes the link to oscillate on its trunnions, thus giving to the link block a further travel than already acquired through


LANGVALVE GEA
S.S.ZENITH CI
23-38-63

the frames. When the block is in the center of link the valve will open the ports only to the amount of the lead; upon movement of the block either way from the center, the port opening becomes larger and the point of cut-off later. It will be plain that one end of the link produces the reversed action of the other end. The throw of the eccentric should usually be about two-thirds of that employed where the Stephenson link is used, but this is capable of variation either way at the hands of the designer.

It will be noted that the eccentric obtains a powerful leverage on the link center, due to the difference in length between the long and short arms of frames, shaft 6 constituting the fulcrum. Similarly, a leverage is obtained with regard to the port opening movement. In this case point 10 becomes the fulcrum, which will be plain if we regard the distance from 10 to 11 as the long end, and from 10 to the center of link block as the short end of a lever. The two levers described constitute a system of compound leverage, which allows unbalanced valves to be handled with ease, and brings a large percentage of the whole wear to the main fulcrum pin 6, where it is easy to provide ample bearings as well as adjustability for wear.

As shown in Fig. 1, the link block is in position for cutting off steam at three-quarters of the stroke (in this case the full gear position), from which it will be apparent that a much shorter link is employed than ordinarily. In an engine of forty-two inches stroke the link need not be longer than from eighteen to twenty inches between the center of the block in full gear ahead and the like point backing, unless a later cut is desired than here supposed. The slip of the link block can be held within very narrow limits if proper care be given to the hanging of the bridle rod 15, as any slipping action comes from this source.

After applying the Lang valve gear (as originally designed) to a number of engines, it was found that absolute rigidity of the main fulcrum pin of the valve gear brought about a slight distortion of the motion wherever there existed any lost motion in the main-shaft bearings; or, in other words, the rising and falling of the main shaft gave to the link a slight switching or hunting

movement, which while not apparent to the eye still existed in a greater or less degree. Besides this, difficulty was found in properly locating the exact point on engine for the main pin, where the plumb line could not be used. To obviate these difficulties the plan was adopted of linking the fulcrum pin to main shaft, thus accurately determining its distance from same, and of further pivotally linking said pin to engine column by means of thrust bar, as seen to the right in the photographs. This arrangement, as will be readily seen, places the entire valve-gear system in synchronism with the main shaft and causes the latter to receive its thrust.

From the illustrations it will show that the fulcrum pin is linked to main shaft by means of a pair of carrier bars (so called), one on each side of eccentric sheave, and having their lower ends expanded into bearings in which the main shaft revolves. The longitudinal space taken up on main shaft is the same as where two eccentrics are used.

In the case of the S. S. *Zenith City* some difficulty was encountered in connecting up the reverse motion, it being obviously impossible to lower the reverse shaft, besides which the movement required was much shorter than previously in use with the Stephenson link motion. The plan resorted to is as follows: Horse heads were thrown back 45 degrees on reverse shaft and there secured; connection was then made downwardly to long arm of reverse bell crank, clearly shown in one of the photographs, the short arm of the latter being then coupled up to radius rod, which arrangement not only gives reduction of reverse movement but also permits of the horse-head screws being manipulated in the ordinary manner.

Concluding, it may be said that the I.P. valve gear on S. S. *Zenith City* drives a large double-ported slide valve, under steam pressure of 60 to 70 pounds, without distress of any kind, although the valve is unbalanced and unprovided with any weight-relieving device.

DATA H.P. VALVE GEAR, S. S. *ZENITH CITY.*

Kind of valve	Piston.
Steam	Outside.

Lap of valve, inches	$1\frac{1}{8}$
Lead (constant), inch	$\frac{3}{16}$
Inner clearance, inch	$\frac{3}{32}$
Diameter of eccentric circle, inches	7

WORKING FULL GEAR.

Travel of valve, inches	$5\frac{1}{4}$
Port opening, top, inch	$\frac{15}{16}$
bottom, inches	$1\frac{1}{8}$
Cut-off top, inches of stroke	$24\frac{1}{2}$
bottom, inches of stroke	$24\frac{5}{8}$
Piston travel for full valve displacement, top, inches of stroke	$5\frac{1}{4}$
bottom, inches of stroke	$4\frac{7}{8}$
Period of pause, top, inches of stroke	$5\frac{1}{2}$
bottom, inches of stroke	5
Release, top, inches of stroke	$34\frac{3}{4}$
bottom, inches of stroke	34
Compression, top, inches of stroke	35
bottom, inches of stroke	$35\frac{3}{4}$
Preadmission, top, inches of stroke	$39\frac{7}{8}$
bottom, inches of stroke	$39\frac{7}{8}$

Working at earlier points of cut-off data approximates closely to that already given for I.P. gear.

VALVE-GEAR DATA, S. S. *ZENITH CITY*.—ENGINE $20\times38\times60$, 40-INCH STROKE—I.P. VALVE GEAR.

Kind of valve	Double-ported slide.
Lap of valve, inches	$1\frac{1}{2}$
Lead, constant, inch	$\frac{1}{4}$
Inner clearance, inch	$\frac{1}{8}$
Diameter of eccentric circle, inches	$6\frac{3}{4}$

WORKING FULL GEAR AHEAD.

Valve travel, inches of stroke	$5\frac{3}{16}$
Point of cut-off, top, inches of stroke	$24\frac{3}{4}$
bottom, inches of stroke	$24\frac{3}{16}$
Piston travel for full valve displacement, top, inches of stroke	$5\frac{1}{8}$
bottom, inches of stroke	5
Period of stroke while valve pauses, top, inches of stroke	$4\frac{1}{2}$
bottom, inches of stroke	4
Release, top, inches of stroke	$34\frac{1}{2}$
bottom, inches of stroke	$34\frac{1}{8}$
Compression, top, inches of stroke	$35\frac{7}{16}$
bottom, inches of stroke	$35\frac{3}{4}$

Preadmission, top, inches of stroke	$39\frac{13}{16}$
bottom, inches of stroke	$39\frac{7}{8}$
Port opening, top, inch of stroke	$\frac{31}{32}$
bottom, inches of stroke	$1\frac{5}{32}$

WORKING WITH HORSE HEAD HALF SCREWED UP.

Valve travel, inches of stroke	$4\frac{3}{8}$
Point of cut off, top, inches of stroke	19
bottom, inches of stroke	$19\frac{5}{16}$
Port opening, top, inch of stroke	$\frac{31}{32}$
bottom, inch of stroke	$\frac{3}{4}$
Piston travel for full valve displacement, top, inches of stroke	$3\frac{1}{2}$
bottom, inches of stroke	3
Period of pause, top, inches of stroke	$3\frac{1}{2}$
bottom, inches of stroke	3
Release, top, inches of stroke	$31\frac{3}{4}$
bottom, inches of stroke	$31\frac{1}{2}$
Compression, top, inches of stroke	33
bottom, inches of stroke	$33\frac{1}{4}$
Preadmission, top, inches of stroke	$39\frac{11}{16}$
bottom, inches of stroke	$39\frac{3}{4}$

WORKING WITH HORSE HEAD SCREWED UP.

Valve travel, inches of stroke	$3\frac{7}{8}$
Point of cut-off, top, inches of stroke	$11\frac{3}{4}$
bottom, inches of stroke	$12\frac{1}{2}$
Port opening, top, inch of stroke	$\frac{31}{32}$
bottom, inch of stroke	$\frac{7}{16}$
Piston travel for full valve displacement, top, inches of stroke	$1\frac{3}{4}$
bottom, inches of stroke	$1\frac{1}{2}$
Period of pause, top, inches of stroke	$1\frac{7}{8}$
bottom, inches of stroke	$1\frac{3}{4}$
Release, top, inches of stroke	28
bottom, inches of stroke	$27\frac{7}{8}$
Compression, top, inches of stroke	$30\frac{1}{4}$
bottom, inches of stroke	30
Preadmission, top, inches of stroke	$39\frac{5}{16}$
bottom, inches of stroke	$39\frac{1}{2}$

I.P. GEAR, BACKING, FULL GEAR.

Valve travel, inches of stroke	5
Port opening, top, inch of stroke	$\frac{7}{8}$
bottom, inches of stroke	$1\frac{1}{8}$
Cut-off, top, inches of stroke	22
bottom, inches of stroke	$22\frac{1}{2}$

Piston travel for full valve displacement, top, inches of stroke	$4\frac{1}{2}$
bottom, inches of stroke	5
Period of pause, top, inches of stroke	$3\frac{1}{2}$
bottom, inches of stroke	4
Release, top, inches of stroke	$33\frac{3}{4}$
bottom, inches of stroke	$32\frac{3}{4}$
Compression, top inches of stroke	34
bottom, inches of stroke	35
Preadmission, top, inches of stroke	$39\frac{1}{4}$
bottom, inches of stroke	$39\frac{11}{16}$

REGARDING EFFECT OF COMPRESSION IN STEAM CYLINDERS ON ECONOMY.

By B. C. Ball, Member.

In the May number of the Journal an article was printed which contained extracts from some letters which passed between Chief Engineer B. F. Isherwood, U. S. Navy, and Professor Dwelshauvers-Dery, of the University of Liège. These letters referred to some tests made by the latter, with a view of determining the effect of compression on the economy of the steam engine.

Mr. Isherwood did not agree with the deductions which Professor Dwelshauvers-Dery made from the tests, and the latter made some subsequent tests, which seemed to indicate that Mr. Isherwood's theories were correct; and Professor Dwelshauvers-Dery was then converted to this theory.

It is unfortunate that more detailed information was not given regarding the conditions of these tests, for it is well known by all who have experimented on the subject that the speed of the engine, the relation of condensation surfaces to the volume of the cylinder, and other conditions, affect very materially the results obtained.

It is quite certain that the degree of compression which would be most economical for a high-speed engine would not necessarily be the most economical one for running at a slower speed.

In Mr. Isherwood's letters he makes some statements which seem to be open to question, and I wish to call attention to two of them, and endeavor to explain my views regarding them.

He states that in an engine in which the admission valve has no lead, it is physically impossible to compress the back pressure in the cylinder to a pressure equal to the initial pressure, be the back pressure and degree of compression what they may. This may be true of engines running at very slow speeds, but it

is absolutely certain that in high-speed engines, at least, it is not true, for I have taken a great many indicator cards in which the compression not only reached the initial pressure but was carried considerably above it, and when the valve opened for admission the pressure dropped to the initial pressure.

This condition is often met with in high-speed compound engines running non-condensing, and in order to obviate this difficulty the exhaust edges of the valve must be cut away considerably, particularly in the low-pressure cylinder.

The main point to which I wish to call attention, however, is the following statement of Mr. Isherwood's theory regarding compression:

"First. No matter what may be the degree of expansion and compression adopted, and no matter what may be the proportion of the dead space at one end of the cylinder to the capacity of the cylinder, there is neither economic gain nor loss by the compression of the back pressure. In all cases the quantity of steam compressed reproduces during the immediately succeeding stroke of the piston exactly the work of that compression.

"Second. Compression does not modify in the slightest degree the injurious effect of the dead space on the economy of the steam. The injurious effect of that space remains exactly the same, whether there be or be not compression, or what may be the degree of compression."

It is not made clear in this statement just what allowance is made for condensation. Reference is made to this important factor in one or two places in the letters, but nothing is said as to how it is brought into the problem, if at all. Probably no one questions the fact that if condensation were eliminated compression to the initial pressure would give the best economy, and the same law would make expansion to the back-pressure line most economical, with no condensation. The laws governing this case are so well known and so easily demonstrated that they need no further attention.

Mr. Isherwood has had such a wide experience in the effect of condensation on the economy of steam engines that he undoubtedly must have made allowance for it in his theory. In

the explanation of his theory Mr. Isherwood imposes the condition of keeping the indicated work constant. The reason for this is not stated, but it is probably his idea that in this way the relation of condensation to indicated work is maintained practically constant.

The greatest trouble about laying down a law for the most economical expansion and compression curves is that these curves will vary widely under different conditions of speed, condensation surfaces, etc. The same engine running at a low speed will give a much higher terminal pressure, with cut-off and other conditions remaining exactly the same, than it would at a higher speed. This, of course, is due to the re-evaporation, during the expansion, of steam condensed at admission.

Under ordinary conditions of piston speed, etc., it has been found that the hyperbolic expansion curve fits the average indicator diagram very closely, and engine designers almost universally use this curve in proportioning their cylinders. Assuming, then, that we are using steam on hyperbolic expansion and compression curves, we can use the laws of the hyperbola without going astray.

The formula for the work done in a cylinder during hyperbolic expansion is $W = p_2 v_2 Log_e r$, in which

W = foot pounds of work;

p_2 = final pressure, in pounds, per square foot;

v_2 = volume of one pound of the gas at terminal pressure;

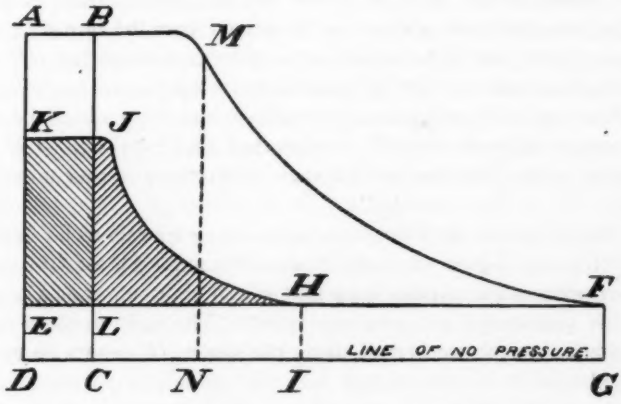
r = ratio of expansion.

According to this formula the amount of work done per pound of gas is absolutely independent of the amount of clearance space, provided always that the compression completely fills the clearance space to initial pressure. An engine with 100 per cent. clearance would be just as economical as one without clearance under these conditions. The reason this is not true in practice is that larger clearance involved larger areas for condensation for the same amount of indicated work, and the same ratio of expansion.

As a consequence of this law, if we were using gas not subject to cylinder condensation the economy of the engine would not

be affected in any way by the amount of clearance, as long as compression was carried to the initial pressure; but inasmuch as we do have condensation in steam engines, we cannot, with economy, expand to the back-pressure line nor compress to initial pressure. It is more economical to throw away some steam by free expansion, at release and into the clearance space, than to utilize the perfect diagram.

The main argument advanced by Mr. Isherwood in support of his theory is that in the compression of steam into the clearance space exactly the same power is absorbed as is given out again by that steam during the next stroke. In one sense this is true, but in the sense in which he uses the argument it is not true, or, at least, is not the whole truth.



Referring to his figure on page 316, of which Fig. 1 is a duplicate, he says that the steam *KJLE*, which is compressed back into the boiler, absorbs the amount of work *JHL* and gives off exactly that amount of work again by expansion.

To show that this is not the whole truth, let us assume that in this cylinder the piston can travel to the line of no clearance *AE*. In this ideal case the area of indicated work which the amount of steam *AM* would give would be *AMFEA*; but as the piston

can only travel to *BL*, the amount of indicated work for this same amount of steam, *AM*, will be *BMFLB*. Therefore, *ABLE* is the area of work lost, due to clearance. Now, if we compress from *H* to *J*, an area of work, *JHL*, will be taken from the diagram to accomplish this compression, and the amount of steam compressed will be represented by *JK*.

This amount of steam, *JK*, working between two hyperboles will develop the area of work *JHEK*. The line *EK* being a hyperbole in its limiting case when it coincides with the axis. Thus it will be seen that if we can use the compressed steam, without loss by free expansion, we will *gain* the amount of work *JLEK*, for by expending the area of work *JHL* we make it possible to get back the amount of work *JHEK*.

It is immaterial how far the line *JK* is located from the line of no clearance *AE*; for it is a well known characteristic of the hyperbola, that two of these curves drawn from the line *KJ* produced, to the line *EF*, will enclose an area exactly equal to *JHEK*, if they cut from the line *KJ* produced, a length equal to *JK*.

The logical conclusion of the above reasoning would be, of course, compression to *B*, thus saving the total area *ABLE*, which is the loss due to clearance when there is no compression.

The fallacy of Mr. Isherwood's reasoning appears to lie in the fact that he argues that the steam *JK* cannot be used in any more economical manner than that which he assumes as a basis of his reasoning—*i. e.*, no compression. The right to place this limitation on the method of using the steam *JK* cannot be conceded.

He could, with equal propriety, say that there is no gain by expansion, for the amount of steam saved by cutting off would do the same work if carried to the end of the stroke that it will do in the next succeeding stroke. This reasoning assumes that there will be no expansion in the next stroke, and the reasoning in the two cases is parallel.

The laws governing cylinder condensation involve so many variables that they are not actually known, and are not likely to be for a long time to come. So many small and seemingly in-

significant conditions change results so greatly that we cannot get an accurate knowledge covering the whole subject.

The best we can do is to experiment along certain lines, and then confine our reasoning to similar conditions. In order to establish by experiment a law regarding compression it would be necessary to conduct series of tests under widely varying conditions of speed, clearance space, etc., then endeavor to lay down a law from these experiments.

The tests of Professor Dwelshauvers-Dery neither prove nor disprove any theory, for, in the first place, the results of these tests vary, in one series over 14 per cent., under the same set of conditions. This wide discrepancy in tests of this character makes them questionable foundation for a law. The speed of the engine during the tests was so slow (48 revolutions per minute) that it would be manifestly unfair to use the results obtained to deduce a law for engines running at all other speeds.

In 1894, Professor Jacobus conducted a series of tests on a high-speed engine at Stevens Institute to determine whether or not compression to initial pressure was as economical as medium compression in that class of engine. The results of the test were reported to the American Society of Mechanical Engineers, and are contained in Volume XV of the Transactions.

The tests showed an advantage of about 10 per cent. in favor of medium compression. From these tests it seems fairly well settled that in high-speed engines, at least, with about 10 per cent. clearance, a medium amount of compression gives appreciably better economy than compression to initial pressure. It would not be safe, however, to reason from this, that all classes of engines, at all speeds, would show the same degree of saving with the same amount of compression.

ON THE COMPRESSION OF BACK-PRESSURE STEAM INTO THE WASTE SPACE OF CYLINDERS.

By Commodore B. F. Isherwood, U. S. Navy.

A REPLY.

The present number of the Journal contains an article by Mr. B. C. Ball, under the title of "Regarding Effect of Compression in Steam Cylinders on Economy," which is a statement of his opinions adversely to the demonstrated conclusions given by me on the same subject and published in the preceding May number of the Journal.

The demonstrations referred to are strictly geometrical and are absolutely irrefutable, as Mr. Ball will discover as soon as he undertakes to substitute mathematical argument for opinion. Of course, I am aware that it is impossible to satisfy mere opinion, but, in this case, the conclusions in question were confirmed by the admirable experiments made on the subject by Professor Dwelshauvers-Dery of the University of Liége. Mr. Ball, however, rejects the experimental proof, evidently without knowing anything about it, as peremptorily as he rejects my mathematical demonstrations, which he seems to comprehend just as little. Neither does he appear to understand that dissent is not refutation.

Mr. Ball complains that Professor Dwelshauvers-Dery has not given enough information regarding the conditions of his experiments. Now the Professor's reports are so full that it is difficult to imagine any data wanting. The Professor is not only an eminent scientist, but a most acute and sagacious experimenter. He has a perfectly equipped laboratory for the purpose, and none knows better than he how to use it. He describes, too, as admirably as he executes. Perhaps Mr. Ball has not seen the Professor's reports published in the "Revue Méchanique";

if he has not, and I can scarcely believe that he has, judging from his remarks, he is not warranted in making groundless objections to them; on his own showing, however, he is as inaccessible to the Professor's experiments as to my mathematics.

Mr. Ball states as his opinion "that the speed of the engine, the relation of condensation surfaces to the volume of the cylinder, and other conditions affect very materially the results obtained."

I assert and will prove that neither the speed of the engine nor the relation of the condensation surfaces to the volume of the cylinder, has the slightest influence on the problem. If the same engine be worked at the same speed of piston with different degrees of compression, with the same measure of expansion for the steam, with the same back pressure against the piston exclusive of the compression, and with the same indicated pressure upon the piston, then, the economic result, as far as the relation of pressures produced to steam expended, will be exactly the same be the degree of compression what it may, or be there no compression. This conclusion is absolutely demonstrated by the reasoning based upon my diagram reproduced in Mr. Ball's article. Neither I nor any other person can add anything to the force of this demonstration, or improve its directness, simplicity and clearness, because the diagram and the demonstration exactly represent the processes of nature for the case in question. I recommend Mr. Ball to study them more carefully than he seems to have done. When he understands them, he will find he has already received an answer to his objections, and much more not to be found elsewhere. If Mr. Ball can show an error in this diagram or in the reasoning based upon it, I ask him to point it out; if he cannot, what is the value of his opinions?

The reason why the reciprocating speed of the piston must be the same in experiments with different degrees of compression, has no relation to the degree of compression. It is necessary from two causes applicable to all experimentation on steam engines made to discover the effect of particular adaptations or variations in the regimen of the steam on the economy of the steam.

The first cause is steam leakage past the valves and piston of the cylinder.

The second cause is the external radiation of heat from the boiler, steam pipe, and cylinder, when the cost of the work is required in units of fuel; and only from the steam pipe and cylinder when that cost is required in units of heat expended to produce the steam used.

The faster the engine is worked with the same regimen of steam, the more economically will the work be done, because both the leakage and the radiation are constant per unit of time, be the reciprocating speed of the piston what it may, while the work done is directly as the speed of the piston. The leakage, however, is only an accident resulting from a mechanically badly conditioned engine, a condition which need not and ought not to exist; and the loss of heat by external radiation can be reduced by proper non heat-conducting cladding to a very small proportion, and although the extent to which they exist (and, strictly speaking, they always exist more or less) causes the faster piston speed to give more economical results than the slower piston speed, yet neither exerts the slightest influence on the problem of compression.

With the same initial pressure, the same measure of expansion, and the same back pressure against the piston exclusive of the compression, the greater the degree of compression employed, the less will the indicated pressure be. Now, for equal piston speed for the same engine, the heat expended produces the mean total pressure on the piston calculated down to the zero of pressure, consequently the more and more the back pressure is compressed, the less and less fraction of this total pressure will the indicated pressure be, and the less and less economically will the indicated pressure be produced. In other words, the mean total pressure being constant, and the indicated pressure being a function of the compression, decreasing as the latter increases, the greater the latter the less will the former be. This loss of economy referred to indicated pressure, however, takes place just the same by reduction of indicated pressure, the back pressure remaining constant, whether the back pressure be compressed or not, or whatever

may be the degree of its compression. The loss in question referred to indicated pressure, results wholly from the fact that the heat expended is proportional to the mean total pressure of which the indicated pressure is a decreasing fraction as the compression increases.

If the economic comparison be made in the case of different degrees of compression, or of no compression, for the mean total pressure on the piston, then the equality of the indicated pressure in the different cases need not be insisted on, and the result would show, evidently, the same economy for all degrees of compression or for no compression.

Mr. Ball will doubtless now perceive the necessity of making comparative experiments with different degrees of compression, or without compression, with the same piston speed always; and, when the economy is to be determined in relation to indicated pressure or power, with the same indicated pressure, and in the same engine, as was done in the faultlessly executed experiments of Professor Dwelshauvers-Dery, who perfectly understood the nature of the problem, the proper conditions required, and the reasons for the same. His experiments were, therefore, made in an unexceptionable manner, and his results are inattackable.

There may be added that the greater the degree of compression employed, the higher, other things equal, must be the initial pressure in the cylinder, in order to obtain, with other things equal, equal indicated pressure.

The compression of the back pressure, be the degree of compression what it may, is absolutely without effect upon the economy, except as regards the two slight incidental effects hereinafter stated, and they, strictly speaking, do not belong to the problem, which is one of the relations of pressure to steam expenditure in each particular case.

The variations of economy, referred to indicated pressure when different indicated pressures are employed, and when the same or different back pressures exist, being caused wholly by the proportion of the indicated pressure to the mean total pressure—a self-evident proposition—compression of the back pressure can have no influence on them. And in the case of an engine work-

ing without leakage and without external radiation of heat, differences in the reciprocating speed of its piston, other things equal, would be equally without effect on the economy.

Mr. Ball states "that the degree of compression which would be most economical for a high-speed engine would not necessarily be the most economical one for running at a slower speed." This statement is without meaning, and shows a hopeless misunderstanding not only of the subject, but ignorance of the behavior and condition of steam in the working cylinder.

Mr. Ball incorrectly quotes me as saying "that in an engine in which the admission valve has no lead it is physically impossible to compress the back pressure in the cylinder equal to the initial pressure, be the back pressure and degree of compression what they may." The inaccuracy consists in the interpolation of the words "in which the admission valve has no lead." I am obliged to notice this inadvertence because, though apparently unimportant, it is really important, as my position is the physical impossibility of compressing the back pressure in the cylinder to the inital pressure, regardless of the setting of the valve, lead or no lead, and without any limitation or qualification.

Mr. Ball adds: "This may be true of engines running at very slow speeds, but it is absolutely certain that in high speed engines, at least, it is not true, for I have taken a great many indicator cards in which the compression not only reached the initial pressure, but was carried considerably above it, and when the valve opened for admission the pressure dropped to the initial pressure."

This statement shows that Mr. Ball has neither read his indicator diagram in a sane manner, nor understood why I required the admission valve in experimenting to be without steam lead. If that valve have steam lead, the diagram cannot show whether the compression ends before the initial pressure is reached or not. If it have no steam lead, the indicator diagram will always show the compression ending before the initial pressure is reached, no matter whether the engine works fast or slow, or what may be the other conditions. With steam lead, the ending of the compression curve is unavoidably masked or obliter-

ated by the lead, which allows boiler steam to enter the cylinder before the stroke of the piston is completed. In this case, the diagram shows no interruption of apparently the compression curve, but the end of that curve is there notwithstanding, only overlain by the pressure due to the lead. Now, set back the eccentric sufficiently, and the hook or crochet marking the end of the compression will appear. There are no exceptions to this fact. Mr. Ball's indicator diagrams were taken from engines whose valves had steam lead, and he did not know that the phenomena of compression could not be observed under that condition. Neither has he understandingly read the paper he attempts to criticise, in which all this is really given, though perhaps too briefly for him, as it was written from one expert to another who did not need such explanations. Professor Dwelshauvers-Dery's experiments were made on an engine whose admission valve not only had no lead, but opened a trifle after the crank had passed its dead point. The accomplished professor perfectly understood his problem and how to solve it. If Mr. Ball had ever taken indicator diagrams from a cylinder in which the back pressure was compressed, and having an admission valve set without lead, he never would have written his article.

Further, Mr. Ball has never seen an indicator diagram, whether the back pressure was compressed or not, that showed a higher pressure at the commencement of the stroke of the piston than the proper initial pressure. Such a fact is a physical impossibility. What he has mistaken for such higher pressure was due, not to any real pressure there, but to the momentum of the vertically moving parts of the indicator, when such parts were violently thrown upwards by the sudden admission of steam under high pressure. This momentum carried these parts necessarily correspondingly above the initial pressure, an appearance found oftener on diagrams from cylinders in which there is no compression of the back pressure. And, as the movement of the piston is continuous from the dead point of the crank, that movement and the time required for the vertically moving parts of the indicator to rise above the initial pressure and fall back to it through the height due to the momentum in question, pro-

duce an area on the diagram apparently representing pressure, though there is really no corresponding pressure there. To take indicator diagrams is one thing, to correctly interpret them is another and very different thing. The one requires hands, the other needs a head.

The reason why the compression pressure can never reach the initial pressure, involves considerations of the conjoint action of the heat, of the temperature, and of the relative mass of the metal of the cylinder and of the steam contained in the cylinder.

In what follows, the supposition is made that the cylinder loses no heat by external radiation, and gains none from the friction of its piston and valves, nor from the contact of its valve chest.

Suppose, further, that in such a cylinder the steam be used without expansion, the pressure from end to end of the stroke of the piston being uniform, and that the back pressure be not compressed. Suppose the metal of the cylinder at the commencement of the stroke of the piston to have the temperature normal to the pressure of the steam within it. Then, why, under these conditions, should the steam lose any heat in passing through the cylinder to the condenser? And if it lose no heat, there will be no liquefaction of it in the cylinder.

To avoid any misunderstanding here, it is proper to state that the heat in the steam in the cylinder is not Regnault's total heat, but a less quantity of heat by the quantity transmuted into the external work done during a stroke of the piston; this latter quantity of heat being supplied in the boiler, and not taken out of the steam in the cylinder. For instance, the heat found in the condenser would be less than Regnault's heat, which is the total heat of generation of the steam under the pressure employed and above the feed water temperature, by the thermal equivalent of the external work done by the steam during its generation.

If steam had the molecular constitution of an ideal gas, it would, under the conditions, lose no heat in its passage through the cylinder into the condenser, because its molecules would have no inter-attraction, but as it is a vapor, its molecules have such an attraction, and as the steam expands when the exhaust

port is opened to the condenser, falling from the cylinder pressure at the end of the stroke of the piston to the condenser pressure, the heat required to be transmuted into the internal work of overcoming its molecular attraction is taken from the steam itself which thus loses that quantity of heat and has its temperature correspondingly decreased. The steam in the cylinder was in the saturated state during the steam stroke of the piston, but, from the instant the exhaust port opened, it became superheated by the difference of heat normal to the cylinder pressure and to the condenser pressure, less the heat transmuted into the internal work of overcoming the mutual attraction of its molecules through the space represented by the difference of its bulks under the two pressures. Now the latter quantity of heat being less than the former quantity, the steam remaining in the cylinder and forming the back pressure there, will still be in the superheated state, but at a less temperature than the temperature of the steam in the cylinder during the steam stroke of the piston, and being in contact with the metal of the cylinder will proportionally reduce the temperature of that metal, so that when the cylinder receives steam for another stroke of its piston, there will be a liquefaction of as much of this entering steam as is necessary to furnish the heat to restore the temperature of the metal of the cylinder to the temperature of the entering steam.

If the steam had no molecular attraction, the temperature of the back pressure in the case above given, would be exactly the same as the temperature of the entering steam, and the temperature of the metal of the cylinder would be constant throughout the double stroke of the piston, and the same as the temperature of the entering steam. In this impossible case, impossible because steam has considerable molecular attraction, compression of the back pressure could be carried to exactly the initial pressure but no farther. In all possible cases, the back pressure cannot be compressed to the initial pressure, and if the steam be used expansively, the more expansively it be used, the lower will be the pressure to which the compression will be carried, other things equal, because the greater will be the difference of temperature between the initial temperature and the back pressure tem-

perature. The non-expansive use of steam allows the maximum compression pressure to be obtained, but it can never reach the initial pressure in the cylinder.

The back pressure steam in the cylinder is, in all cases, whether the steam be used with or without expansion, or with what measure of expansion it may, in the superheated state, but with a temperature less than the temperature of the steam entering the cylinder, the back pressure being greatly less than the pressure of that steam. The heat corresponding to the difference between these two temperatures, into the mass of the metal of the cylinder and its adjuncts, into the mean specific heat of the metal between these temperatures, will be taken from the latent heat of the entering steam, a sufficient quantity of which will be liquefied to furnish it. The water of this liquefaction is revaporized during the exhaust stroke of the piston under the lessened pressure of the back pressure, and at the expense of the heat in the metal of the cylinder. In the case of using steam expansively, there is no revaporization of this water of liquefaction during the expansion portion of the stroke of the piston. All of it is revaporized during the exhaust stroke of the piston. The steam corresponding to the water of liquefaction in question, is first evaporated in the boiler. On entering the cylinder it is liquefied, imparting its latent heat to the metal of the cylinder, and raising that metal to its own temperature. The liquefaction in the cylinder is completed never later than the closing of the cut valve when the steam is used expansively, and probably it always takes place in the waste space at one end of the cylinder, before the commencement of the stroke of the piston. The water of liquefaction always remains in the cylinder as such until the exhaust port opens, when it is revaporized to the condenser by the heat imparted to the metal of the cylinder during the liquefaction of the steam whose liquefaction produced it in the cylinder at about the commencement of the stroke of the piston. The heat thus taken out of the metal of the cylinder during the portion of the exhaust stroke between the commencement of that stroke and the point in that stroke at which compression commences, if there be compression, is exactly equal to

the heat imparted by the liquefying steam to the metal of the cylinder previous to the closing of the cut-off valve. This heat is thus simply transferred from the boiler to the condenser through the intermediary of the metal of the cylinder and without doing work in its passage. In quantity per double stroke of piston, it is represented by the weight of steam liquefied into the latent heat of that steam.

There is no revaporization of any of the water of this liquefaction during the expansion portion of the stroke of the piston, notwithstanding the continuous fall of pressure due to the expansion, because the temperature of the metal of the cylinder continuously falls during the continuous expansion of the steam, and the temperature of the water of liquefaction in contact with that metal falls with the temperature of the metal, the temperature being less than normal to the pressure of the expanding steam considered to be in the saturated state. The pressure of this expanding steam is really greater than the normal pressure due to the temperature of the metal of the cylinder, because it is superheated, as will be shown later, and has the additional pressure due to the superheating. On the opening of the exhaust port, the pressure falls to the condenser pressure and the water of liquefaction is immediately revaporized, as that pressure is much less than normal to the temperature of the metal of the cylinder.

Now the facts to be kept in evidence as regards the problem of the compression of the back pressure in a cylinder using the steam without expansion are these:

That the temperature of the metal of the cylinder during the exhaust stroke of the piston is less than its temperature during the steam stroke of the piston.

That the back pressure steam is in the superheated state, while the steam in the cylinder during the steam stroke of the piston is in the saturated state.

And, that the temperature of the metal of the cylinder and of the saturated steam was the same during the steam stroke of the piston; while the temperature of the metal of the cylinder and of the superheated back pressure was the same during the

exhaust stroke of the piston; because, in each case, the steam was in contact with the metal of the cylinder, and the two had, therefore, to have a common temperature.

Lastly, that the product of the mass of the cylinder, including its adjuncts, by the number of degrees of temperature it may pass through, into the mean specific heat of its metal for that range of temperature, compared with the product of the liquefied mass of steam contained in the cylinder into its latent heat, is so very great, that the changes in the temperature of the metal of the cylinder produced by the absorption of any heat taken out of the steam, are so small that they may be neglected without causing error in a general presentation of the subject though they would have to be included in any particular case requiring exact numerical results. In fact, for all the cases of practice, and whether the steam be used with or without expansion, the temperature of the metal of the cylinder is almost constant, the changes being limited to a very few degrees. The temperature of the metal of the cylinder will lie between the temperature of the entering steam and a few degrees lower, the fall being greater as the steam is used more expansively, but the mean temperature of the metal will vary, even in extreme cases, only slightly from the extremes.

Now, let the exhaust port be closed before the piston reaches the end of its exhaust stroke, the superheated back pressure will then be compressed, and its temperature would be increased by the thermal equivalent of the work of compression, were it not in contact with the metal of the cylinder at the temperature of the back pressure at the commencement of its compression. The consequence is that the heat of compression is immediately taken up by the metal of the cylinder, the temperature of the compressed back pressure remaining throughout its compression sensibly at the temperature it had before its compression began. Hence, the compressed back pressure is a case of the compression of a gas (as the back pressure is always superheated) at constant temperature, and the curve of pressures thus described will be the Mariotte curve, which is experimentally found to be the fact.

Of course, the temperature of the metal of the cylinder would be increased by this thermal equivalent; but very slightly, owing to the enormous difference between the quantity of heat generated by the compression, and the quantity of heat required to raise the temperature of the metal of the cylinder and its adjuncts a fraction of a degree. The compression, therefore, takes place upon a surface which is practically at a constant temperature during the compression.

Now, as the compression proceeds, the resulting pressure of the compressed back pressure becomes greater and greater, while the degree at superheating of that back pressure becomes less and less, until the superheating is entirely lost which happens when the temperature of the compressed back pressure becomes normal to its pressure as saturated steam. This is the point at which the compression ceases, for after that point is attained the compressed back pressure liquefies by further compression. It is physically impossible for the curve of compression to be carried above this point. The movement of the piston past this point indeed does work upon the compressed back pressure, the thermal equivalent of which work is imparted to the back pressure steam and tends to raise its temperature, but, as the compression would be effected upon a surface cooler than the compressed back pressure, this rise of temperature is simultaneously taken out of the back pressure steam, whose point of liquefaction thus remains unchanged, the thermal equivalent referred to having passed into the metal of the cylinder. Consequently, the line of back pressure produced by the further movement of the piston between the point of liquefaction in question and the end of the stroke of the piston, will be horizontal, and such is experimentally found to be the fact.

Thus, in the case of using steam without expansion and with compression, the steam is of uniform pressure and is in the saturated state from the beginning to the end of the steam stroke of the piston; then, from the beginning of the exhaust stroke of the piston to the point in that stroke at which the compression begins, the back pressure steam is in the superheated state with a constant degree of superheating.

From the point in the exhaust stroke of the piston at which compression of the back pressure begins, to the point at which liquefaction of the compressed back pressure begins, the back pressure continuously increases in pressure and continuously decreases in its degree of superheating, until it becomes saturated, when the pressure of the compressed back pressure and the temperature of the metallic surface upon which it is compressed become normal to each other as saturated steam, a conjunction that necessarily happens before the initial pressure is reached, after which conjunction the back pressure steam continues in the saturated state to the end of the exhaust stroke of the piston and at a constant pressure.

The greater the difference between the temperature of the saturated steam entering the cylinder and the temperature of the superheated back pressure when its compression begins, the lower will be the point of liquefaction of the compressed back pressure steam, and the lower, consequently, will be the limit of pressure possible to be attained by the compression of the back pressure.

The cessation of the compression is due to the fact that the temperature of the compressed back pressure has become that which is normal to its pressure as saturated steam. Hence, the lower the temperature of the metal of the cylinder on which the compression is effected, the sooner will the pressure meet its normal temperature and liquefaction commence. And, as the temperature of the metal of the cylinder at the end of the exhaust stroke of the piston before the entrance of the boiler steam for the return steam stroke of the piston is always lower than the temperature of the entering steam, the compression pressure can never equal the initial pressure.

When the steam is used expansively, the steam in the cylinder between the commencement of the stroke of the piston and the closing of the cut-off valve is in the saturated state, a sufficient quantity of the steam as it enters the cylinder being liquefied to furnish the latent heat required to raise the temperature of the metal of the cylinder from its temperature at the end of the exhaust stroke of the piston to the temperature of the en-

tering steam. But from the instant that expansion commences, the expanding steam is necessarily superheated, and the degree of superheating increases as the expansion pressure decreases.

The reason of this superheating is that the metal of the cylinder is at the temperature of the initial pressure when the expansion commences. As the expanding steam does both internal and external work as it expands, it would be in the saturated state throughout the entire expansion did it not receive heat from the metal of the cylinder—an external source—and, if in the saturated state, its temperature would be less than the temperature of the metal of the cylinder with which it was in contact, and less and less as the expansion progressed farther and farther. Now as the temperature of the expanding steam in the saturated state is less than the temperature of the metal of the cylinder with which it is in contact, it must necessarily become superheated to the temperature of that metal, abstracting from the metal the requisite quantity of heat for the purpose, and correspondingly reducing the temperature of the metal. But, as has already been explained, the enormous difference between the quantity of heat in the metal of the cylinder and its adjuncts, relatively to the quantity of latent heat in the steam per stroke of piston, causes the fall of temperature of the metal of the cylinder due to the expansion of the steam to be correspondingly small, so that, practically, the temperature of the metal of the cylinder remains nearly constant throughout the expansion, which, in turn, causes the temperature of the expanding steam to be also nearly constant; consequently, the case of the expanding steam is that of a natural gas expanding at constant temperature, a result of which is that the expansion curve is the Mariotte curve very nearly, a fact confirmed by every indicator diagram ever taken from the cylinders of steam engines using steam expansively. This expansion curve could not be produced in any other manner than the manner just described, and it is an irrefutable experimental proof of the accuracy of the reasoning throughout this paper.

The coincidence of the expansion curve and of the Mariotte curve has often been remarked of late years, but the cause has

never before been stated. Slight discrepancies will be found between these curves, as the expansion curve is given on an indicator diagram, but if the imperfections of that instrument be taken into account, nearly all of them will disappear.

The heat taken out of the metal of the cylinder to superheat the expanding steam, will be greater with greater measures of expansion, and it has to be restored to the metal at the commencement of the succeeding steam stroke of the piston by the liquefaction of the requisite quantity of entering steam to furnish it. The more expansively the steam is used, the greater, therefore, is the difference between the temperature of the metal of the cylinder at the end of the exhaust stroke of the piston and of the entering steam, and the greater, consequently, is the steam liquefaction in the cylinder, and the loss of useful effect by the steam. The greater, also, is the difference between the initial pressure in the cylinder and the highest pressure of the compression curve, when the back pressure is compressed.

In the case, then, of using steam expansively with compression of back pressure, the steam in the cylinder from the commencement of the stroke of the piston to the closing of the cut-off valve is in the saturated state. From the closing of the cut-off valve to the end of the stroke of the piston, and thence to the point in the compression curve of the back pressure at which liquefaction takes place, the steam is superheated. And from that liquefaction point to the end of the exhaust stroke of the piston, the compressed back pressure is in the saturated state. Only from the pressure of the steam at the closing of the cut-off valve, can the weight of steam received by the cylinder from the boiler, exclusive of the weight liquefied at the commencement of the stroke of the piston, be calculated. And only from the pressure of the compressed back pressure at its point of liquefaction, and the distance of this point from the end of the exhaust stroke of the piston, can the weight of back pressure steam compressed into the waste space be calculated.

Although, rigorously speaking, there is neither gain nor loss of pressure on the indicator diagram, relatively to weight of steam expended as shown on the diagram, by compressing the

back pressure, there are, incidental to this compression, two small economies of fuel. One is that the heat equivalent of the work of compressing the back pressure passes into the metal of the cylinder, thereby correspondingly increasing its temperature and lessening the weight of entering steam required to be liquefied at the commencement of the stroke of the piston. The other is due to the purely practical consideration that by either steam lead, or by compression, the piston, piston rod, crosshead, and connecting rod must be brought gradually to a state of rest at the end of the stroke of the piston, and then started from a state of rest in the opposite direction without shock or concussion. Theoretically, this is done perfectly by the crank, supposing the various articulations of the heavy moving parts to have no "play" or slackness of fitting, but, as in practice there must be some play or slackness allowed to permit the influx of the lubricant and to prevent the heating which results from the excessive friction due to tight fittings, the crank movement must be supplemented by a gradual increase of back pressure against the piston as the latter approaches the end of its stroke, so as to gradually change the sides of the journals at which the slackness exists before the piston reaches the end of its stroke. The shock or concussion takes place, if at all, exactly at the dead point of the crank, and at the instant when the direction of the piston's movement is changed. The back pressure against the piston by either steam lead or by compression, must be, when the piston reaches the end of its stroke, greater than the steam pressure on the piston when it reaches the end of its stroke, both pressures being counted above the normal back pressure due to the condenser pressure.

Now the effect of steam lead is not mechanically as good as the effect of compression for this purpose, because it acts more suddenly; nor is it economically as good, because the increase of back pressure by lead is wholly an economic loss, whereas its increase by compression is no loss at all, so that the substitution of the latter for the former causes an economic gain practically equal to the back pressure due to the lead.

In practice, no matter what may be the degree of compression

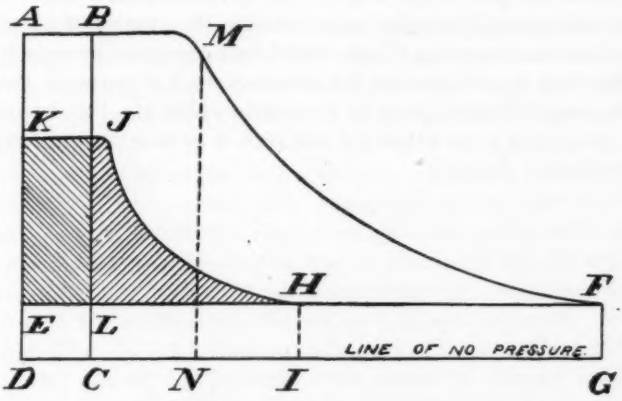
employed, some steam lead is always found necessary. The reason is that as the compression curve can never be carried up to the initial pressure in the cylinder, the difference must be supplemented by steam lead. The universal employment of steam lead, even with the highest degree of compression used, is an experimental proof that the maximum compression pressure is below, and generally much below, the initial steam pressure in the cylinder.

I will state here, and for the first time in the literature of steam engineering, that the sole cause of the liquefaction of steam in working cylinders—cylinder condensation as it is usually termed—is the expansive use of the steam. When the steam is used without expansion there will be no steam liquefaction in the cylinder, but just in proportion to the measure of expansion with which the steam is used, will the liquefaction be. A pound weight of steam of a given pressure used with a given measure of expansion, will produce the same amount of steam liquefaction in a cylinder, be the dimensions and proportions of the cylinder what they may, and be the reciprocating speed of the piston what it may. This I regard as the most important generalization ever made in relation to the behavior of steam in working cylinders. That a part of the steam entering a working cylinder was liquefied therein, has been known as an unexplained fact from the days of James Watt: the true explanation—the cause—is now given for the first time, namely, the superheating of the expanding steam at the expense of heat in the metal of the cylinder, which heat has to be restored to the metal from the latent heat of the entering steam at the succeeding stroke of the piston, no work being at any time obtained from this heat.

The same facts, also, combined with other causes, furnish the proper explanation of the failure of expanding steam to give anything like the economy predicable of it. The more the stsam is expanded, the greater is its liquefaction in the cylinder, the case being one of difference between a gain and a loss inseparable from the gain. The water of liquefaction not being

revaporized during the expansion of the steam no work is derived from the heat lost by the liquefaction.

I have only time and space enough left to notice Mr. Ball's method of showing that the loss of useful effect by clearance or waste space at the end of the cylinder, may be reduced to nothing by compression, however great the space may be.



He supposes, referring to the reproduction of my diagram in his article, that the piston can travel to the line of no clearance *AL* notwithstanding compression. This supposes a physical impossibility, for with compression there must always be some clearance, some space to compress into. Only without compression can the piston be supposed to work without clearance. I quote Mr. Ball as follows:

"Now if we compress from *H* to *J*, an area of work *JHL* will be taken from the diagram to accomplish this compression, and the amount of steam compressed will be represented by *JK*.

"The amount of steam *JK*, working between two hyperboles will develop the area of work *JLEK*. Thus it will be seen that if we can use the compressed steam, without loss by free expansion, we will *gain* the amount of work *JLEK*, for by expending the area of work *JHL* we make it possible to get back the amount of work *JHEK*."

In this amazing argument Mr. Ball does not perceive what stares him in the face from the diagram, that to obtain the area of work *JHEK* by carrying the stroke of the piston to the end of the clearance or through the clearance, there must be expended exactly the amount of work *JHEK* and not the lesser amount *JHL*. This error vitiates his entire argument and makes his result of the gain of the work *JLEK* an obvious absurdity. His argument properly made proves exactly the reverse of his conclusion, and shows, as I have stated, that compression—much or little—has no influence on the injurious effect of the waste space. The proof of this is given in the article which Mr. Ball attempts to criticize, but he either did not read it or failed to appreciate its force.

NOTES.

ZINC AS A PROTECTIVE COATING FOR IRON AND STEEL.

The "Engineer's Gazette" of June, 1900, contains an abstract of a paper recently presented at a meeting of the Cleveland Institution of Engineers held at Middlesbrough, England, by Mr. Sherard Cowper-Coles, A. M., Inst. C. E., M. I. M. E., Westminster.

The author remarked that zinc as a protective coating for iron, after a trial of over fifty years, has proved itself to be cheaper and more effective than painting. Iron coated with zinc by passing it through a molten bath of that metal has the somewhat misleading name of 'galvanized iron;' this would seem to infer that the coating was effected by a galvanic current. It is intended to apply, however, to the nature of the protective influence and not to the process. Two metals in contact in the presence of moisture form a galvanic couple, and the result of the galvanic action set up under these conditions is analogous to that which takes place in a galvanic battery, viz: the electro-positive element is dissolved or corroded, while the electro-negative element remains unaffected or protected so long as the action continues, the intensity of the chemical action being governed by the relative position of the two metals in the electro-chemical series.

Tin and iron, and copper and iron, standing some distance apart, give rise to a decided galvanic action, in which the iron, as the electro-positive metal, is corroded more rapidly if the coating is partly removed than it would be if not coated. With zinc and iron the positions are reversed. Zinc is positive to iron, and, consequently, whatever galvanic action is set up by their contact in the presence of moisture or other electrolytes must have for effect the solution or corrosion of the zinc and the protection of the iron. As compared with tinning, galvanizing is a compara-

tively new industry, having been worked in Europe since the early part of last century. At the present time it has assumed the proportions of a very large industry, the value of the shipments amounting to over £2,000,000 per annum. Unlike tinned iron, galvanized iron resists the action of the weather to a remarkable degree. Zinc surfaces after brief exposure become coated with a thin film of oxides which adheres tenaciously, forming an insoluble protective coating to the underlying zinc; so long, therefore, as the zinc surface remains intact, the iron is effectively protected from corrosive action.

In America lead or terne plates are largely used for roofing, and when kept well painted are found to last a considerable time. The great advantage of lead-coated sheet is its greater pliability and the fact that it does not diminish the strength of the iron and is very adhesive, thus enabling lead-coated wire or sheet to be bent and seamed, which is difficult with galvanized sheet, due to the brittleness of the iron and the tendency for the zinc to flake off. The first method employed for coating iron with zinc was electrolytic, but it was soon abandoned in favor of the dry or molten process, as the results obtained were not satisfactory and the process was found to be too slow and costly. The zinc put on by the hot process is uneven, and the increased weight is often a matter for consideration; the thickness can only be varied within narrow limits, the minimum weight of zinc being about 1½ ounces per square foot, maximum, 3 ounces, except when the molten zinc can be raised many degrees above the most economical working temperature and the articles removed gradually from the bath so as to give the zinc time to drain off before getting chilled, or in the case of sheet or wires which are capable of having some of the zinc squeezed or wiped off by rollers or other suitable mechanical means.

The Heathfield process of galvanizing consists of drawing the sheets out between two sets of rollers. The approximate cost of a hot-galvanizing plant for 2,400 tons of galvanized corrugated iron and 100 tons gutters and ridging per annum would be £1,500, and to double this production would require £500 more. The amount of zinc to charge the above-named plant

would be about 25 tons. Hot galvanizers experience considerable trouble and loss from the formation of an alloy of iron and zinc in the galvanizing bath, commonly known as dross. The percentage of iron in the alloy varies from 5 to 10 per cent., and, the fusing point and specific gravity of the alloy being higher than that of zinc, the dross has to be removed from time to time by means of iron ladles. The iron dross usually amounts to about 25 per cent. of the total amount of zinc used. Small quantities of tin are sometimes added to the bath to improve the brightness of the deposit and increase the size of the spangles, but this attractive appearance is gained at the cost of the durability of the protection afforded to the underlying metal, as the spangles render the coating unequal in thickness and inferior in quality; for these reasons some of our leading engineers now specify that galvanized sheets shall be without spangles.

The author next dealt at length on the disadvantages of hot galvanizing. He stated that both pickling and hot galvanizing reduce the strength of the wires, and distort and render brittle iron and steel of small section. Most manufacturers have ceased to galvanize the best class of steel wire (which costs some £35 per ton) on account of the great risk of rendering it worthless. This is unfortunate, for corrosion is found to be very marked on the inner strands of wire rope, and the advisability of protecting the steel is unquestionable. Various attempts have been made to reestablish cold or electro-galvanizing, but these have not been successful, as the process was found too slow and costly, and the zinc coating porous. Zinc flashing had been adopted by some, and found to be very useful for the preservation of steel work during transport. The process of electro-zincing was then described at length, and illustrated by diagrams of the plant. The first care, the author stated, is to have a properly-prepared surface; all traces of grease and rust must be removed. Machined work, which is usually very greasy, should be immersed for about half an hour before pickling in hot solution of caustic soda, about two pounds to one gallon of water. The work should then be placed in a hot pickle containing about 1 per cent. of commercial sulphuric acid. If in pickling the sur-

faces of iron are carefully cleaned from the oxide by dipping in acid, or removed by means of the sand blast, it is found the corrosion is always much lessened.

The removal of the mill scale from forgings and plates has always been a matter of considerable difficulty, the scale in many places being about $\frac{1}{32}$ inch in thickness, and strongly adherent. The usual practice is to place the iron in a solution containing one part of hydrochloric or sulphuric acid to ten parts of water for a period varying from half an hour to twenty-four hours. The Admiralty specify that all steel steam pipes, boiler and collector tubes, and all plates for boilers are to be pickled in a liquid consisting of nineteen parts of water and one of hydrochloric acid until the black oxide or scale formed during the manufacture is completely removed. During the process of pickling, large quantities of magnetic oxide and scale become detached from the plates, which, if allowed to remain in the pickle to be further acted on by the acid, would form a serious form of loss. To overcome the magnetic scale, a collector has been devised. A description of this was given and its advantages enumerated. In America an electrical pickling process has been tried, the solution used being sulphate of soda or sulphuric acid. The plates to be cleansed form the negative electrode, and an iron plate the positive. The hydrogen set free from the surface of the plates reduces the oxide on the surface, and decomposes any grease that may be present. A voltaic method is stated to do the work with more certainty, and with less danger to the surface of the metal in the way of pitting. The bath consists of dilute sulphuric acid containing a depolarizer such as nitric or chromic acid.

Dealing with the effect of pickling and rusting on the strength of iron and steel, the author stated that the bending tests with wrought-iron joists showed substantially the same for all conditions. The maximum loads carried showed differences, being highest in the natural condition and lowest when freshly pickled. Rusting also produces a diminution of strength more apparent in the galvanized articles than in those not galvanized. Some recent experiments in regard to the decrease in the strength of

iron after being pickled and galvanized are very interesting. A dozen eye-bolts, all precisely alike, were carefully selected. Six were laid on one side and the other six sent to be galvanized. Afterwards one of each of the ungalvanized bolts was connected to each of the galvanized ones. They were placed between two pieces of iron and the nuts screwed up until the eye-bolts broke. Invariably it was found that it was the galvanized eye-bolts that were the ones to break; not in one instance did the ungalvanized ones give way.

Boatbuilders state that galvanized iron is "rotten"; where great strength is required they select the iron and have the fittings forged by a good smith, but do not have them galvanized. Galvanized iron is frequently used in places where little strength is required, but where rust would be objectionable. For this purpose galvanized iron leaves nothing to be desired, but where the full strength of the material is required and rust stains are objectionable, the object may be attained by cleaning them by the sand-blast process, and afterwards coating them by the cold galvanizing process; both these methods are now largely used for the above-named purpose.

Sand blasting is often used instead of pickling, and for many purposes it is found to give better results, especially for cast-iron work, from which it is very difficult to remove the last traces of acid. A description and drawings of a complete sand-blast equipment were given, as well as of the apparatus for galvanizing for the electrical process. A special feature of this process is that plates of very large dimensions can be speedily zinced on one or both sides, which is a matter of some importance, now that plates of great length are being put on the straight run of steamers' hulls. The frames of such craft as torpedo-boat destroyers can also be galvanized after riveting up. The regenerative electro-zincing process was largely used, and the electrolyte employed is constituted as follows: Zinc sulphate, 35 ounces; sulphuric acid, $\frac{1}{10}$ ounce; water, 1 gallon. The method of regenerating the electrolyte and of testing the acidity of it was described; also a new method for the rapid valuation of zinc dust.

It was stated that among the advantages of the patent regenerative electro-zincing process were increased rate of deposit; nonformation of dark, spongy patches on the work; no reduction of the tensile strength of steel of high-breaking load; girders and other similar built-up structures, propeller shafts, etc., can be coated without difficulty in whole or in part; there is no waste of materials or expense incurred in the renewal of the bath; all the zinc is consumed, no dross being formed; the zinc coating is very adhesive and elastic and comparatively free from impurities; the cost of plant is less than a hot-galvanizing plant of equal capacity; the plant can be worked intermittently without any waste of materials, or allowed to stand for months, thus rendering it economical for manufacturers to have their own galvanizing plants.

The success of this process has rendered possible the coating of wire, tubes and other articles which hitherto was impracticable, owing to the great rapidity at which they had to be zinced to obtain commercial results. The zinc can be deposited to any thickness for the purpose of coating ironwork to be erected in inaccessible exposed positions. Zinc dust is used, and it is 20 per cent. cheaper than virgin spelter. The author described the plant required for electro-zincing wire, tubes, plate, etc., and he cited a number of figures comparing the cost of ordinary electro-zincing with that of the Cowper-Coles regenerative zincing process; for the former it was 0.21d. per square foot, in the latter 0.15d. The Cowper-Coles electro-zincing plant, erected at the Imperial German dockyard at Wilhelmshaven, also the regenerative electro-zincing plant of the Germania Shipbuilding and Engineering Company at Tegel, near Berlin, were described, also the plant erected at the works of Messrs. Ross and Walpole Limited, at Dublin.

CHANGES IN LLOYD'S RULES FOR SHAFTING.

Changes in the rules for determining the dimensions of shafting were recently decided upon by Lloyd's Register and are

now in force. The amended paragraphs are referred to by their numbers and are as follows:

Shafts.—30. All shafts are to be made of good material and are to be examined when rough turned and when finished. In the case of screw shafts, scrap steel is not to be used. It is recommended that these be made of ingot steel or forged from blooms made from rolled iron bar of good, fibrous quality.

31. Gages of an approved description for testing the truth of the crank shafts are to be supplied with all new engines, and adjusted in the presence of the surveyor.

32. The length of the stern bush is to be at least four diameters of the shaft. It is recommended that the shaft liner should be continuous the whole length of the stern tube, and that the after end should be tapered in thickness and made watertight in the propeller boss. If the liner is made in two pieces the joint should be burned. If the liner does not fit tightly at the part between the bearings in the stern tube, the space between the shaft and the liner should be charged or "forced" with a plastic material, insoluble in water and non-corrosive. If two liners are used, it is recommended that they be tapered in thickness at the ends, and that the shaft should be lapped or protected between the liners. In this case, and also if no liners are used, the diameter of the shaft should be $\frac{21}{20}$ of that required for a shaft with a continuous liner.

For dimensions of shafts, see the formula in paragraph 59.

Rules for Determining Sizes of Shafts.—59. The diameters of intermediate shafts are to be not less than those given by the following formula:

For compound engines with two cranks at right angles:
Diameter of intermediate shaft in inches

$$= (.04A + .006D + .02S) \times \sqrt[3]{P}.$$

For triple-expansion engines with three cranks at equal angles:
Diameter of intermediate shaft in inches

$$= (.038A + .009B + .002D + .0165S) \times \sqrt[3]{P}.$$

For quadruple-expansion engines with two cranks at right angles:

Diameter of intermediate shaft in inches

$$= (.034A + .011B + .004C + .0014D + .016S) \times \sqrt[3]{P}.$$

For quadruple-expansion engines with three cranks:
Diameter of intermediate shaft in inches

$$= (.028A + .014B + .006C + .0017D + .015S) \times \sqrt[3]{P}.$$

For quadruple-expansion engines with four cranks:
Diameter of intermediate shaft in inches

$$= (.033A + .01B + .004C + .0013D + .0155S) \times \sqrt[3]{P}.$$

where

A = diameter of high-pressure cylinder in inches,
B = diameter of first intermediate cylinder in inches,
C = diameter of second intermediate cylinder in inches,
D = diameter of low-pressure cylinder in inches,
S = stroke of pistons in inches,
P = boiler pressure above atmosphere in pounds per square inch.

60. The diameter of crank shaft and of thrust shaft under the collars, to be at least $\frac{21}{20}$ of that of the intermediate shaft. The diameter of thrust shaft may be tapered off at each end to the same size as that of the intermediate shaft.

61. The diameter of the screw shaft to be equal to the diameter of intermediate shaft (found as above) multiplied by $\left(.63 + \frac{.03P}{T}\right)$, but in no case to be less than $1.07T$, where P is the diameter of propeller, and T the diameter of intermediate shaft, both in inches.

This size of screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in paragraph 32. If no liners are used, or if two separate liners are used, the diameter of the shaft should be $\frac{21}{20}$ that given above.

The diameter of screw shaft is to be tapered off at the forward end to the size of the crank shaft.

62. NOTE.—The rules are intended to apply to two-cylinder compound engines, in which the ratio of areas of low and high-

pressure cylinders does not exceed 4.5 to 1; the triple-expansion engines in which it does not exceed 9 to 1; in quadruple-expansion engines in which it does not exceed 12 to 1; and in all cases, as regards the stroke, in which the length of stroke is not less than one-half the diameter or greater than the diameter of the low-pressure cylinder. Engines of extreme proportions beyond these limits being specially submitted to be dealt with on their merits.

Periodical Surveys.—65. The stern shaft is to be examined annually and drawn at intervals of not more than two years.*

These new rules are the result of a long-continued investigation made by the technical committee of Lloyd's.—"Marine Engineering."

ELECTRICAL EQUIPMENT OF SHIPS OF WAR.

In an interesting editorial in the London "Engineering" (May 11, 1900), the present unsatisfactory condition of electrical equipment of war ships relative to that which is desired and hoped for in view of the general and rapid application of electrical appliances elsewhere, is dwelt upon at some length. We quote the following extracts illustrating the principal lines of advance upon which so many engineers are devoting serious thought:

"It is nearly a quarter of a century since the first electric projector was installed on a British ship of war, and yet, so slow is mechanical and scientific progress even in this rapid age, it is still doubtful what design of projector is the best to adopt for naval purposes. In regard to all other departments of a vessel in which electricity might possibly play a part, with the exceptions of incandescent lighting and the working of ventilating fans, there is even more uncertainty in the minds of naval officers and constructors. Experiments are being tried in a tentative fashion in our own and other navies, particularly in the latter; but electrical engineers find it very uphill work to impart their

* On the application of owners, the committee will be prepared to give consideration to the circumstances of any special case.

own enthusiasm to those who have to use the apparatus they design.

* * * * *

"The immense difference between the conditions, of peace and war was brought out recently in the discussion before the Institution of Electrical Engineers of a paper on 'The Electrical Equipment of Ships of War,' by Mr. C. E. Grove, the electrical engineer to the Thames Shipbuilding and Engineering Company, Limited. The author described the hand-regulated lamp of the British Admiralty, and also the motor-controlled lamp used by some foreign nations. Mr. R. E. Crompton took this part of the subject as the text of a severe lecture which he administered to the Admiralty on their want of enterprize, and their inability to appreciate the improvements which scientific men placed at their service, until they were obliged to do so in consequence of the action of other Powers.

* * * * *

"In the matter of communication, again, the means which act well when all is quiet, fail entirely in the din of battle. Our vessels are fitted with voice tubes, 2 inches in diameter; but when the guns are being fired, and, indeed, with less violent noises than are produced by artillery, messages cannot be received by them. Loud-speaking telephones are also being tried in our own and several other navies; but it is reported that in the battle of Santiago the Americans found these to be inaudible. As our ships are equipped at present, there is the certainty that messengers will have to be largely employed to transmit orders during an engagement. The position of a captain within a conning tower will then become very difficult, for until he finds his orders being executed, he will never be certain that the messenger has not been killed before he could deliver it. Mr. Grove suggests the use of a system of printing telegraphs, something after the pattern used by the Exchange Telegraph Company, printing in ordinary letters, as the best method of communicating orders. As 20 to 30 words can be transmitted per minute, and as in addition many orders can be coded or abbreviated, this would furnish a safe and fairly expeditious means of dis-

tributing orders, although it is to be doubted whether it would always keep pace with the captain's requirements.

"Means for working projectors and for conveying intelligence in a ship are, however, matters of minor interest to the electrical engineer. The subject which attracts him is the distribution of power to various parts of the vessel, and he looks forward to the time when the energy required for the whole of the very numerous machines will be supplied by the dynamos. A first-class battleship carries nearly 100 sets of machinery of various kinds, and a first-class cruiser 50 to 70 sets. With the exception of the guns, all were formerly worked by steam, and most of them are still so worked; although in all new ships the electric motor is finding extended application. The most advanced example in this respect is the United States battleship *Kearsarge*, just completed, which carries 55 motors, and a generating plant of 350 kilowatts capacity. If this proves a success, it should give a great stimulus to electric driving on board ship, and bring forward for decision the question whether electricity shall not be used in all positions where it is possible. No one would attempt to argue that steam and hydraulic power do not serve their present purposes in the auxiliary machinery of a ship of war, extremely well. Indeed, one may go further and state positively that electric motors will neither raise the boats and anchors nor rotate the hoists more satisfactorily than is done at present. But it by no means follows on that account that the present engines and rams will be superseded in the near future. They are likely to share the experience of the tank boiler. That was a most reliable apparatus when fairly used, and had a dozen merits which do not pertain to the water-tube boiler. Yet it has been superseded in all the navies of the world, not because it was unreliable, but because a ship is an engine of war, and must be judged as a whole. Speaking generally, no part of a ship is what its designer would desire if he had a free hand. The armor is not thick enough, the guns are not powerful enough, the ammunition supply is scanty, the speed is too low, and the coal supply insufficient. In every particular the authorities have to content themselves with something far short of perfec-

tion, for the simple reason that a ship would not float if they put into it all the arms and appliances which they would like to see there. It is just this consideration which, sooner or later, will lead to the use of electric motors in all the auxiliary machinery of a ship. Their employment will save weight, and the displacement thus rendered available can be applied to other purposes.

"Electric motors are not lighter than steam engines of corresponding power; indeed, when the generators and their engines are taken into consideration they are distinctly heavier. But they use far less coal than do the small engines they replace, and the saving they effect amounts to a large weight in a week. Mr. Grove calculates that the total motor capacity required on a battleship would be 1,885 electrical horsepower, including duplicates, or 1,385 horsepower, excluding duplicates. It is impossible to conceive that all this could be operative at the same moment, even in action. Mr. Grove, therefore, puts the maximum demand at a given instant, including 50 kilowatts for lighting the ship and working the searchlights, at 600 kilowatts. He would provide 800 kilowatts in generators and their engines, and these with steam and exhaust connection would weigh about 80 tons. The weight of the motors would be about that of the engines they displaced, if the saving that would be effected by the substitution of electrical machinery for hydraulic appliances for gun working be reckoned in. Against the weight of the generators and their engines is to be placed that of the main hydraulic equipment of about 25 tons which would be saved. There would also be a slight saving in the weight of the distributing cables and switchboards as compared with the auxiliary steam and exhaust pipes now used. Approximately 55 tons of additional weight would have to be carried, and this can only be economically done if a proportionate saving in coal can be effected.

"It is very difficult to say what amount of coal is used by the auxiliary machinery on board ship, but undoubtedly it is very great, the condensation in the steam pipes alone accounting for a great amount of fuel. Sir John Durston, in his paper on the trials of the *Diadem* (see 'Engineering,' Vol. LXV, page 444,

April 8, 1898), stated that to run one main feed pump, one blowing engine, one auxiliary circulator, one electric-light engine, and two distillery pumps consumed coal at the rate of 6.1 tons per day, when two extreme forward boilers were used, or 3.88 tons when two extreme after boilers were employed. The same machines, with the addition of two evaporators, working compound, consumed 8.8 tons of coal per day when supplied by the forward boilers, and 7.09 tons when supplied by the after boilers. That is, the length of steam pipe between the forward and after boilers accounted for 2.12 tons of coal per day in the first trial, and for 1.71 tons in the second. What the loss is when really long lengths of steam pipe to the forward and after capstans, and to the steering gear, are in operation has never been determined as far as we know, but evidently it must amount to several tons of coal per day. Again, it has been found that in the Japanese ironclad *Shikishima* it takes from $3\frac{1}{2}$ to 5 tons of coal to run for 18 hours a day an engine of 65 indicated horse-power situated 160 feet away from the boiler, the total range of steam pipe connected being 500 feet to 600 feet. This works out to from 7 pounds to 10 pounds of coal per indicated horse-power, of which more than half must have been used in keeping the pipes warm.

"The consideration of such figures as these, somewhat rough though they may be, endorses the belief that the auxiliary machinery of a battleship, steaming at one-fifth power, accounts for about one-fourth of the coal consumed, say 20 tons a day. Sir John Durston, in his paper before the Institution of Naval Architects (see 'Engineering,' March 31, 1899), put the figure at 22 per cent. when the main engines were giving 3,600 horse-power, and this was on a trial in which everything was working at its best. It seems, therefore, reasonable to hope that something like 10 tons of coal might be saved every day a ship was at sea if all the auxiliary machinery could be driven electrically. Thus, by adding 55 tons of machinery to the ship, the coal consumption would be reduced 10 tons a day, or about 12 per cent. at economical speeds. In other words, some 200 tons of displacement would be rendered available for other purposes, such

as increased speed, greater radius of action, heavier armament, or thicker armor.

The time is not yet ripe for such a radical change in the equipment of our ships of war; but there is no doubt that the change will come in the near future. It is, however, one of those alterations of which the great benefit cannot be reaped until it is adopted entirely. It is only when a clean sweep is made of long steam pipes, and a properly designed dynamo room, second in importance only to the main engine rooms, is provided for in the original design, that full economy will be attained.

"The greatest obstacle to progress at present is the steering engine. It is a very bad steam eater, and we are not yet in sight of a satisfactory plan of operating it by electricity. The regulation has to be so minute and so absolute, and withal capable of being effected from so many parts of the ship, that the problem bristles with difficulties. Again, the subject of gun training by electricity is far from simple. M. Canet has attained a considerable degree of success, but to do that he had first to carefully balance his turrets so that they moved with far less than the accustomed efforts. He was fortunate that he could do this, because he occupied an important position in the company which constructed the guns. Most inventors would find that the interests of the firms supplying the artillery did not lie in the direction of introducing refinements to eliminate friction. Nevertheless, in spite of difficulties of many kinds, substantial progress is being made, and each year sees electricity playing a more important part on ships of war. But, just as it is rapidly displacing line shafting and belts on shore, so it is destined to take the place of long steam pipes and hydraulic mains on naval vessels, until some day naval engineers and officers will wonder how they endured the heat and mess of the present system, while constructors will thankfully accept the displacement thus rendered available for other purposes."

QUICK MOBILIZATION OF A SQUADRON.

Just before 4 o'clock on Tuesday afternoon, June 5th, 1900, Secretary Long issued orders assigning the *Indiana* and *Massa-*

chusetts (then lying in reserve, with skeleton crews only, at League Island, Pa.), to the North Atlantic squadron directing that they proceed to Newport with as little delay as possible. The Bureau of Navigation issued the following explanation of the emergency order.

"The commandant at League Island has been ordered to prepare the *Indiana* and *Massachusetts* immediately for a two months' cruise. When ready they will fill with coal at Hampton Roads and thence proceed to Newport, when they will join the North Atlantic squadron. The fitting out is ordered to be done with dispatch, and the Department has directed the vessels at Newport by the 13th, if practicable."

Secretary Long sent out the order just in time to catch the officers and clerks at closing up hour of the business of the day. The order went to every department at Washington, to all receiving ships and navy yards of the northern Atlantic coast, and to the marine barracks. Thence it went to the commanding officers of the ships in reserve and to other ships that were to transfer men to them. In a few minutes the entire naval force within the range of the orders was transformed from a body of men just ready to go ashore after a day's work in office and on shipboard to the same number of hustling men, each one bent on making the utmost effort to promptly carry out the emergency instructions from Washington. The wording of the orders could hardly have been more emphatic if an enemy's fleet was off the coast waiting to attack the city and the safety of the country depended upon the haste with which these two fighting ships could get out to sea to head them off. That was part of the scheme. As it happens that this is the first time the quick-mobilization idea has been carried out in this country, it is little wonder that folks in and out of the service wondered seriously for a few hours if it really did not have a deeper meaning than a drill on a large scale. And even now there are some who refuse to be convinced that it does not mean something more than the naval officers in high places of responsibility will admit.

The *Indiana* and *Massachusetts* had on board when the orders came about 150 enlisted men and a dozen officers each.

But each had well-filled bunkers of coal, magazines and shell rooms filled with plenty of ammunition for guns, and stores enough to permit the ships to go to sea the minute the service complements reported on board. The additional officers and men began arriving the following morning. Officers who would have laughed at you the day before if you had suggested to them that they would be going to sea soon, suddenly received telegraphic orders that said: "Go with all the speed of the railway trains." They left their desks in navy yards and Washington bureaus, dumped collars and changes of underclothes into hand bags on top of their uniforms, grasped swords and went, stopping only long enough to say good-by to their families, if they lived near enough. The next morning they were reporting for duty to the commanding officers of the *Indiana* and *Massachusetts* and taking up their share of the bustling work they found going on board the ships.

At the same time, on board the receiving ships at the New York and Boston navy yards and in the marine barracks there, on board the receiving ship *Richmond* and throughout the entire navy yard at League Island, on the receiving ship *Franklin* at Norfolk, and elsewhere, men and stores were being assembled with a rapidity and certainty that was surprising, except to those who had much confidence in the naval system of doing things quickly and efficiently. On Wednesday morning the men began coming on board the ships, those nearest at hand first, of course. Admiral Sampson's telegram from the Boston navy yard was the first word received from outside the League Island navy yard announcing the departure of men for the ships, and the draft from the *Wabash* arrived that evening with the draft from the receiving ship *Vermont* at the New York navy yard, and were put on board the ships immediately. Before breakfast the following morning, the old, historic *Hartford*, just arrived under the same emergency orders from Norfolk, transferred 140 men to each of the ships, these men being westerners with a liking for the navy which the department has undertaken to satisfy, with good results to the service and the men as well. The first detail of marines came from the Boston barracks, but

they were closely followed by forty men from New York, making a picturesque detail in the scene of hasty work, with their red-faced capes thrown back over shoulders as they marched on board the tug and a few minutes later went on board the ships, then waiting at anchor in the Delaware River, off the League Island navy yard, for the word that would set them steaming down the 100 miles of fresh water to the ocean.

At 8 o'clock on the morning of Thursday, Captain Dickins reported the *Indiana* ready in every way for sea, just forty hours from the time the first order was given. The *Massachusetts* was reported by Captain Train as ready about the same time. During the afternoon Captain Dickins, as senior officer, received his orders from Rear Admiral Casey, commandant of the League Island yard, to proceed to Hampton Roads. At 5.30 o'clock, the tide then serving, the ships got under way and proceeded down the river. At 8.30 o'clock they anchored at the bar to await the morning tide, getting under way again at 5 o'clock on Friday morning and proceeding down the river, dropping pilots off the capes of Delaware, and then on to Hampton Roads. Meanwhile preparations had been made by Rear Admiral Barker, commandant of the Norfolk navy yard, for the rapid coaling of the ships. Although the ships had well-filled bunkers on leaving League Island, a battleship's furnaces eat up a deal of coal, and for a cruise of any length it is necessary to fill up to the limit, which means about 1,200 tons in the case of these two ships. So on arrival at Hampton Roads everything was found ready for immediate coaling. The colliers *Marcellus* and *Lebanon* and coal barges were there and were run alongside the ships by navy-yard tugs without delay.

The *Indiana* and *Massachusetts* arrived at Hampton Roads at 6.30 o'clock Saturday morning and sailed for Newport at 11.30 o'clock Saturday night. The record of the quick coaling of battleships was broken by the two battleships which took on in about 16 hours probably 1,000 tons of coal, and at daybreak were fifty miles up the coast.

From this hasty review it will be seen that remarkably rapid work has been performed by every department concerned. It

was expected that three days would be necessary to get the ships from reserve into active service, but it has been shown that it can be done in two-thirds that time, thanks to the easy working of the naval machinery. Another thing has been demonstrated at the same time, and that is the superiority of the system of laying ships up in reserve instead of putting them in ordinary, or out of commission. People in general scarcely know the difference in the terms in ordinary and in reserve, but there is a great deal of difference. A ship in ordinary is laid up in a navy yard with nobody aboard but one watchman, whose only duty is to look out for fires and trespassers. Navy yard workmen go aboard from time to time and keep guns and machinery and other fittings in as good condition as they can under the circumstances. A ship in reserve, on the other hand, has a small number of officers and a skeleton crew, who live on board and keep the ship in thorough condition for service at any time the orders may come. The skeleton crew then expands readily by the addition of new men into a full complement, and the ship goes to sea with little delay.

The *Indiana* and the *Massachusetts* were the first ships of their class and tonnage to be placed in reserve, and hence this was the first opportunity to try the mobilization idea in haste, as is done by first-class naval powers on occasions, to insure the constant readiness so essential for an efficient navy.—" Marine Review."

THE TAYLOR-WHITE TOOL-STEEL PROCESS OF THE BETHLEHEM STEEL CO.

The Bethlehem Steel Co. recently extended an invitation to the various technical papers to send representatives to its shops at Bethlehem, Pa., on July 31, for the purpose of inspecting the results obtained in actual work with a new kind of tool-steel. Some half a dozen journals, including " Engineering News," responded by sending representatives who were most cordially received and entertained by the officers of the company.

The process above mentioned is the joint invention of Mr. F. W. Taylor, Consulting Engineer, and Mr. Maunsel White, En-

gineer of Tests of the company. Some years ago Mr. Taylor was retained to reorganize the shop methods in vogue at Bethlehem, with a view to greatly increasing the output with the existing equipment. It was found that the forge shop had outrun the machine shop to such an extent that it appeared that the latter, costing a million dollars or so, would have to be duplicated.

Mr. Taylor first sought to systematize the management of the shop and to increase the output per unit, with the aim of ultimately placing all work on the piecework system. It was found that one of the first requisites was a perfectly uniform and dependable quality of tool steel, as with it only could be determined the maximum possible performance of a man or a machine. Investigations were accordingly taken up with the view of producing such a grade of steel. These researches have culminated in the perfection of a process of tempering and hardening tools that gives results greatly superior to any heretofore attained both in uniformity and efficiency, as is demonstrated in the shops of the company.

The new steel may be used at cutting speeds two to four times greater than those possible with other steels. The main lines of shafting of the shops have been speeded up from 90 to 250 revolutions per minute, and the gain in the average amount of metal cut per hour throughout the shop is shown in the following table:

Average.	Oct. 25, 1898.	May 11, 1899.	Jan. 15, 1900.	Gain in cut of 3d over—	
				2d.	1st.
				P. ct.	*P. ct.*
Cutting speed, per min., feet and inches.....	8–11	21–9	25–3	16	183
Depth of cut, inches.....	0.23	0.278	0.30	8	30
Feed, inches	0.07	0.0657	0.087	32	24
Metal removed per hour, pounds	31.18	81.52	137.3	68	340

On the day of the visit above mentioned several tests were made to show to the visitors the superiority of the new steel over other brands, Mushet steel being taken as a representative of the

latter. A heavy cut was taken on a very hard piece of cast-iron for fifteen minutes with a special-process tool, which showed no signs of wear, being only slightly discolored at the point. A tool of Mushet steel on being substituted lasted only twenty-two seconds, the speed, feed and depth of cut remaining the same. A piece of steel shafting was placed in the lathe and run at a cutting speed of 150 feet per minute. The tool became red at the tip but maintained a very good cut. On being removed it still showed a good edge, being only a little hollowed out by the escaping chip on the upper side. It was replaced by a Mushet tool, which lasted only two revolutions, at the end of which the nose was completely worn away.

Ordinary tool steel will not stand a temperature above 600 degrees or 700 degrees Fahrenheit, while the new steel may be worked to a temperature between 1,100 degrees and 1,200 degrees Fahrenheit. When working without water and at the maximum rate upon steel the chips oxidize to a deep blue immediately upon leaving the tool. This enables the foreman to know at a glance whether or not a machine is being worked to its full capacity. These blue chips were to be seen throughout the shop, where the new steel is used exclusively on all varieties of work.

As to physical properties, the improved steel is said to be slightly higher in tensile strength and elastic limit, and to be somewhat softer than ordinary steels. The latter property excludes it from use in making finishing cuts.

Although patents are pending, the method of manufacture is a secret, which the officers of the company declined to reveal. The new process is said to greatly improve the steels now in use, but its effect is most marked on a special steel, the composition of which is also withheld. The improved process undoubtedly consists in exact methods and improved appliances for heating, hardening and tempering.—"Engineering News and American Railway Journal."

A NEW ALLOY.

An alloy of magnesium and aluminum, to which the name of "magnalium" has been given, is attracting some attention on

account of its combination of lightness and hardness, together with other useful qualities. Says "The Mining and Metallurgical Journal" (July 15):

"One profitable use for it is found in the mounting of photographic lenses. Magnalium has a very high reflective power, being about equal to silver in this respect and much above the other speculum compositions.

"Magnalium reflects almost without any absorption, and mirrors made from it are superior to those of glass. Magnalium, being composed of an alloy of magnesium and aluminum, is lighter than glass, and much more so than speculum alloys which are three times as heavy as glass. These are but two applications of the new alloy on which experiments are being constantly made, and which promise to be remarkably useful. It is unaffected by air and water, and withstands the oxygen acids to a high degree, but is attacked by the alkalies. In specific gravity it is less than aluminum, which is 2.7, ranging from 2 to 2.2, and can be worked extremely well in the lathe. Its melting point lies between 600 degrees and 700 degrees centigrade, depending upon the proportion of the magnesium. Increasing the amount of this metal the alloy increases in hardness and can be rolled or drawn out into wire. Various properties are obtained with different amounts of magnesium. If with 100 parts of aluminum 10 parts by weight of magnesium are incorporated, the resulting alloy has the working qualities of rolled zinc; with 15 parts of magnesium, those of cast brass; with 20 parts it resembles soft copper, and with 25 parts, hard copper. The metal as now made contains from 10 to 15 parts of magnesium to 100 parts of aluminum, and has a smooth and even surface. On it may be cut the small screw threads necessary in optical work, and it is so hard and firm that a piece of sheet aluminum may be cut with a magnalium knife."

SHIPS.

UNITED STATES.

West Virginia, Nebraska, California, Maryland, Colorado and **South Dakota.**—A circular defining the chief characteristics of three armored cruisers authorized by act of Congress, approved March 3, 1899, and of three armored cruisers, authorized by act of Congress approved June 7, 1900, was issued, July 31, by the Navy Department, and from this the following extracts are taken:

* * * * *

1. In the act of Congress making appropriations for the naval service for the fiscal year ending June 30, 1900, approved March 3, 1899, provision is made under "Increase of the Navy," for *three armored cruisers of about 12,000 tons trial displacement*, carrying the heaviest armor and most powerful ordnance for vessels of their class, to be sheathed and coppered, and to have the highest practicable speed and great radius of action, and to cost, exclusive of armor and armament, not exceeding $4,000,000 each.

2. In the act of Congress approved June 7, 1900, making appropriations for the naval service for the fiscal year ending June 30, 1901, the following provision is made, under "Increase of the Navy," viz:

"That for the purpose of further increasing the naval establishment of the United States, the President is hereby authorized to have constructed by contract, * * * three armored cruisers of about thirteen thousand tons trial displacement, carrying the heaviest armor and most powerful ordnance for vessels of their class, and to have the highest practicable speed and great radius of action, and to cost, exclusive of armor and armament, not exceeding four million two hundred and fifty thousand dollars each." * * *

3. It will be observed that this circular provides for six armored cruisers, authorized by two acts of Congress, viz:

"Three under the act of March 3, 1899, to be *sheathed and coppered*, and

"Three under the act of June 7, 1900, for which no provision is made for sheathing and coppering."

4. Two classes of bids are desired on the armored cruisers authorized by the act of March 3, 1899, namely, bids for sheathed and coppered vessels, in accordance with the requirements of the act, and bids for the same ships not sheathed and coppered, the Department reserving the right to adopt either form of construction, in case it should be authorized to do so, at the price named in the bid for such sheathed or unsheathed ships, respectively, the award, however, being based on the bid for sheathed and coppered ships.

5. Bids on the armored cruisers authorized by the act of June 7, 1900, should be on unsheathed vessels only.

7. Bids must state specifically whether they are for sheathed or unsheathed vessels.

8. It will be observed that the acts of March 3, 1899, and June 7, 1900, authorizing the construction of the six armored cruisers herein described provide as follows:

"Act of March 3, 1899.—That not more than *two* of the *three* armored cruisers authorized by this Act shall be built in one yard or by one contracting party, and that not more than *one* shall be built on the Pacific Coast.

"Act of June 7, 1900.—That not more than *two* of the *eight* vessels authorized by this Act, namely, the three armored cruisers described in this circular, two battleships and three protected cruisers, shall be built in one yard or by one contracting party, and that not more than *two* of the said *eight* vessels shall be built on the Pacific Coast."

9. To carry out the provisions of the above-quoted acts, the Department, in advertising for these vessels to be built by contract, will invite proposals as follows:

Class I.—For the construction of the hull and machinery of each of said vessels, including engines, boilers and their appurtenances and spare parts, and for their equipment complete in all respects, except as hereinafter stated, and

for the installation of ordnance and ordnance outfit, in accordance with plans and specifications provided by the Secretary of the Navy.

Class II.—For the construction of the hull and machinery of each of said vessels, including engines, boilers, and their appurtenances and spare parts, and for their equipment complete in all respects, except as hereinafter stated, and for the installation of ordnance and ordnance outfit, in accordance with plans and specifications submitted by the bidder.

10. In order that ample time may be given to all ship and marine engine builders who may desire to submit their own plans of the hull and machinery, or of either, this circular, defining the general requirements of the Department for the vessels referred to, is issued. The general plans may be examined in the bureaus on and after October 8, 1900. The plans will be ready for distribution among prospective bidders on application November 8, and the bids will be opened at the Department on December 7.

* * * * *

17. The Department will reserve the right to reject any and all bids not considered advantageous to the Government, and no bids will be considered which propose to furnish vessels of less than 13,400 tons trial displacement for the unsheathed vessels or of less than 13,800 tons trial displacement for the sheathed vessels, or of less than 22 knots speed on trial, or having a bunker capacity of less than 2,000 tons of coal.

18. Trial displacement is to be understood as meaning the vessel and her equipment complete, with 900 tons of coal, plus two-thirds of ammunition, plus two-thirds of total stores and provisions, crew, and 75 tons of fresh water for steaming purposes.

19. Bids must state specifically the price for each vessel.

20. Each vessel is to be built, fitted, and equipped complete for sea by the contractor, with the exception of sea stores, ordnance, and ordnance outfit, and such other articles as are hereinafter enumerated as to be supplied by the Government.

21. The contractors will fit and secure in place all armor, ordnance, and ordnance outfit.

23. The maximum time allowed for completion will be limited by the Department to thirty-six months for each vessel, and no bids will be considered which propose to exceed that limit of time. Failure to complete the vessel or vessels within the time specified in the contracts will involve penalties as follows, viz: $300 per day for the first month succeeding the expiration of the period fixed by the contract, and $600 per day thereafter until the vessel is completed and delivered as prescribed by the contract.

24. The speed trial of the proposed vessels will be conducted after the vessels are fully completed, in accordance with the terms of the contracts, subject to conditions prescribed by the Department, and if on trial the average speed shall equal or exceed a speed at sea of 22 knots per hour for four consecutive hours the vessels will be accepted, in so far as the speed is concerned.

25. If the speed falls below 22 knots and exceeds 20 knots per hour the vessel will be accepted, so far as speed is concerned, at a reduced price, the reduction being at the rate of $50,000 a quarter knot deficiency of speed from 22 to 21½ knots, and at the rate of $100,000 a quarter knot deficiency of speed from 21½ to 20 knots.

26. If the speed falls below 20 knots per hour the vessel will, in the discretion of the Secretary of the Navy, be rejected, or accepted at a reduced price to be agreed upon between the Secretary and the contractor. In case of rejection any money that may have been paid the contractor on account shall be refunded.

27. The final trial of the vessels will not take place until a period of six months has elapsed from the date of preliminary acceptance.

29. The vessels are to be driven by twin screws.

30. The hull is to be of steel, having a double bottom and close water-tight subdivisions.

31. The arrangement of decks above water is to be such as will provide ample freeboard and good berthing accommodations, the

after body of the vessels being carried to the same height as the forward body.

32. There will be two steel military masts, fitted with upper and lower fighting tops, and with search-light platforms, signal yards, and gaffs.

33. It is contemplated to supply the following boats, viz:

One 50-foot steam cutter, one 36-foot steam cutter, two 36-foot launches, four 30-foot cutters, two 30-foot whaleboats, one 30-foot gig whaleboat, one 30-foot barge, two 20 foot dinghies, one 16-foot dinghy, two 12-foot punts, two 18-foot Carley life rafts; all of which must be stowed so as to be as far as practicable free from injury from the blasts of the guns.

34. The armored protection to the hull is to be as follows:

a. A water-line belt 7 feet 6 inches in width, extending the entire length of the vessel. Said belt, for a distance of about 244 feet abreast of the engines and boilers, to consist of armor 6 inches in thickness at the upper edge and maintaining this thickness downward for a distance of 4½ feet, from which point it tapers to 5 inches at its lower edge. The belt is to be completed to the bow and stern by armor plates having a uniform thickness of 3½ inches.

b. Above the main belt, the side will be protected by armor of 5 inches in thickness for a length of about 232 feet, extending vertically to the main deck.

c. Transverse armor 4 inches in thickness to be worked in at the ends of this 5-inch side armor, thus forming a central casemate for ten 6-inch guns.

d. The four 6-inch guns on the upper deck at the corners of the superstructure to be protected by armor of 5 inches in thickness.

e. The armor for the 8-inch turrets will be 6 inches in thickness, except the port plates, which will be 6½ inches in thickness.

f. The tops of the 8-inch turrets will be of nickel-steel 1½ inches in thickness.

g. The top of the conning tower will be of nickel-steel 2 inches in thickness.

h. The barbettes for the 8-inch turrets will have a uniform thickness of 6 inches. The ammunition tubes for the 8-inch turrets will be 3 inches in thickness.

i. The conning tower will be 9 inches in thickness, and its tube 5 inches in thickness.

j. The signal tower will be 5 inches in thickness.

35. The main armor belt is to be so placed that its upper edge will be at least 12 inches above the water line at maximum deep-load draught.

36. In addition to the armored protection, the 6-inch guns on the gun deck are to be further protected by splinter bulkheads of 2½-inch nickel-steel between each pair of guns.

37. Wherever armor backing is required it will be of East India teak wood.

38. All the above-mentioned armor and the regular armor bolts therefor, except as elsewhere noted, will be supplied by the Government, and will be fitted in place by the ship contractor.

39. The protective deck, splinter bulkheads, and armor shelves will be of nickel-steel, and will be supplied and fitted by the ship contractor.

41. A protective deck of nickel-steel is to extend throughout the vessel, and will be 1½ inches thick on the flat and 4 inches thick on the slopes.

42. A cellulose belt, about 3 feet thick, of approved material, is to be worked along the sides above the protective deck the entire length of the vessel. The cellulose is to be supplied by the contractor, and shall be fireproofed by a process satisfactory to the Department. The cofferdams will be packed by the contractor.

43. No woodwork is to be incorporated in the hull or fittings, except such as is indispensable.

56. Approved means shall be provided for opening and closing by power, as well as by hand, such water-tight doors as may be designated, with suitable devices for indicating at the operating station whether they are open or closed.

58. Each vessel is to be fitted for a flagship, and ample provi-

sion must be made for the accommodation of the full complement of officers and crew, which will comprise—

1 flag officer,
1 commanding officer,
1 chief of staff,
20 wardroom officers,
12 junior officers,
10 warrant officers,
777 crew, including marines.

822

69. There shall be suitable separate cabins for the flag officer and for the commanding officer. Mess rooms for the wardroom, junior, and warrant officers, and the following staterooms will be required, in addition to those in the cabins, viz:

For wardroom officers, 20.

For junior officers, 12.

For warrant officers, 10.

Also one, with toilet room, for chief of staff.

71. The following machine tools of approved make and design will be furnished and installed by the contractor in the general workshop, with proper line and counter shafting (the former to be run by electric motor), and all necessary and usual spare parts and tools, viz:

One screw-cutting back-geared gap lathe, to swing 30 inches over the ways and to take 10 feet between centers.

One 14-inch screw-cutting back-geared lathe, to take 4 feet between centers.

One column-shaping machine, 26-inch stroke, 26-inch traverse.

One double-geared drilling machine, to drill up to $1\frac{1}{2}$ inches, 18 inches from edge of work.

One 16-inch sensitive drill.

One No. 1 Universal milling machine.

One combined hand punch and shears, with 6-inch shear blades, to cut $\frac{3}{4}$-inch round iron and $\frac{3}{8}$-inch steel plate, and to punch $\frac{3}{8}$-inch holes in $\frac{3}{8}$-inch plate.

One emery grinder, two 12 by 2 inch wheels.

One 30-inch grindstone.

Six bench vises.

72. A suitable place will be provided for a blacksmith shop, suitable tools for which will be furnished and installed by the contractor.

74. The following auxiliary machinery of approved make and design, in addition to that pertaining to the main engines and their dependencies, is to be supplied by the contractor; is to be operated by steam power, and is to be installed complete with all the necessary steam and exhaust piping and fittings, viz:

Steam steering engine.

Anchor engine and capstan.

Ash-hoisting engines in each fire room.

Dense-air ice plant of approved type, with a cooling effect of 3 tons of ice per day.

The evaporating plant to consist of four equal units, each having a capacity of 5,750 gallons of fresh water per day; also a distilling apparatus capable of condensing at least 10,000 gallons of water per day.

Five deck winches of 30 horsepower each; each winch to have two hoisting drums, either of which may be operated independently of the other.

Engines of fifty horsepower for each of the heavy boat cranes.

75. The following auxiliary machinery is to be supplied and installed by the contractor, and is to be operated by electric power:

Turret-turning gear.

Ammunition hoists, except those in turrets.

Blowers for hull and for turret ventilation.

76. The following auxiliary machinery will be supplied by the Government and installed by the contractor, and is to be electrically driven:

Turret ammunition hoists.

Rammers for heavy guns.

Elevating gear for heavy guns.

Air compressors for charging torpedo flasks.

77. The machinery in the general workshop will be electrically driven, the contractor supplying the motors therefor.

78. All electric motors and the controlling devices therefor will be supplied by the contractor, except for the torpedo air compressors.

89. The (main) engines will be of the vertical, twin-screw, four-cylinder, triple-expansion type, of a combined I.H.P. of 23,000. The steam pressure at the engines will be 250 pounds. The stroke will be 4 feet. The ratio of cylinders will be at least 1 to 7.30, and the diameters will be sufficient for the required I.H.P. at about 120 revolutions per minute. Each engine will be located in a separate watertight compartment. They will be provided with all the necessary auxiliaries and accessories in accordance with the latest practice of the Bureau of Steam Engineering.

90. There will be thirty boilers of the straight water-tube type, placed in eight watertight compartments. They will have at least 1,590 square feet of grate and about 68,000 square feet of heating surface, and must be able to furnish steam for the main engines and all the necessary auxiliary machinery and other steam machinery throughout the ship with an average air pressure in the ash pits of not more than one inch of water. All the necessary auxiliaries and accessories will be provided for the efficient working of the boilers.

91. There will be four funnels, each 100 feet in height above the base line.

92. One hundred and fifty tons of fresh water will be carried in the double bottom or reserve tanks for the use of the water-tube boilers.

93. The vessel is to be heated throughout with steam, and the steam-heating system will be provided and installed by the contractor.

94. The weight of all machinery and tools, stores, and spare parts will be about 2,060 tons. This weight must include all articles, irrespective of name or use, coming under the cognizance of the Bureau of Steam Engineering, including water in

boilers, condensers, piping, etc., but excluding the reserve feed water in the double bottom or tanks.

95. The armament will be as follows:

Main Battery: 4 8-inch breech-loading rifles of 45 calibers in length.
14 6-inch breech-loading rapid-fire rifles of 50 calibers in length.

Secondary Battery: 18 3-inch breech-loading rifles (14-pounders).
12 3-pounder guns.
4 1-pounder automatic guns.
4 1-pounder single-shot guns.
2 3-inch field guns.
2 machine guns.
6 automatic guns of small caliber.

---

2 submerged torpedo tubes.

96. Suitable and well supported platforms must be provided for all the guns of the main and secondary batteries.

97. The above battery complete will be furnished by the Government, and will be mounted as follows:

a. Four 8-inch guns, in pairs, in two electrically-controlled, elliptical, balanced turrets having inclined port-plates; one forward and one aft, on the line of the keel, and having an arc of fire of 270 degrees.

b. On the upper deck at the corners of the superstructure four 6-inch guns will be mounted in sponsons, one at each corner, each having an arc of fire of not less than 145 degrees.

c. On the main or gun deck ten 6-inch guns in broadside will be mounted, five on each side, having an arc of fire of not less than 110 degrees; that is, 55 degrees forward of and abaft the beam, except the forward pair, which are to be sponsoned and also fire directly ahead.

d. The secondary battery will be mounted in the most commanding positions, having as large an arc of fire as possible.

e. The automatic 1-pounders to be mounted in the lower military tops, and two single-shot 1-pounders in each upper military top.

98. Arrangements must be made whereby all the guns which do not train fore and aft, or nearly so, can be quickly and conveniently dismounted, housed and secured, so as to leave the side practically clear; all dismounting gear to be supplied by the contractor.

115. There will be two submerged torpedo tubes located in one compartment forward. These tubes, complete, with sluice valves, will be supplied by the Government, but will be installed by the contractor, who will also supply the hull castings to which the tubes attach. The torpedo room will also be fitted with proper racks and cages for the stowage of six Whitehead 16.4-foot torpedoes, and convenient means are to be provided for the handling of such torpedoes by means of trolleys, etc. A magazine for the stowage of war heads, to be known as the "torpedo-head room" will be fitted as near the torpedo manipulating room as practicable. A torpedo-firing station will be fitted vertically over each tube, or very nearly so, and will probably be on the deck above the torpedo room. Each station will be fitted with armored protection for the torpedo operator sufficient to shelter him from fire from 6-pounder and smaller guns, and will have a proper port in the side of the ship for use of the torpedo director, and will have a firing circuit to the tube below, and appropriate means of communication with the torpedo room and the conning tower.

116. The air compressors for charging the topedoes and the accumulators and separators will be supplied by the Government, but will be installed by the contractor, who will also supply and install such piping as may be necessary, or, if electric power be used, the Government will supply the electric motors, controllers, and compressors, and the contractor will install the same, and will supply and install the necessary conductors.

117. The total weight of the armament and the ordnance outfit, including the torpedo outfit, is 407.5 tons.

118. The total weight of the ammunition and ordnance stores is 432 tons.

119. The total ordnance weights are 839.5 tons.

120. Ordnance weights to be carried on trial displacement are:

	Tons.
Guns, mounts, etc.,	407.5
Two-thirds ammunition and stores,	288.0
Total,	695.5

123. The electric-generating plant will consist of seven units, each unit to consist of an engine and dynamo mounted on a combination bedplate. Three units shall have a rated output of 1,250 ampères each at 80 volts, and four units shall have a rated output of 625 ampères each at 80 volts. The total weight of the seven units complete must not exceed 141,000 pounds. The total weight of the whole electric installation, including dynamos, engines, bedplates, all fittings, wiring, tools, stores, instruments, and six search lights, must not exceed 158.7 tons.

124. The total weight of the equipment outfit complete must not exceed 305.83 tons.

The ships to be completely wired by the contractor for electric light and power, with all necessary fixtures, fittings, dynamos, engines, bedplates, wiring, instruments, tools, spare parts for engines, dynamos, etc., electrical stores, and search lights.

126. The following articles will be furnished by the Government:

Boats and their spars.

Blocks.

Portable furniture.

Mess tables and benches.

Cooperage.

Anchors, chains, cables, and all other equipment articles enumerated in paragraph 125*a* of this circular.

Ordnance and ordnance outfits of all kinds.

Stores of all kinds, except as specially mentioned.

127. General dimensions, etc., of vessels, Department's design:

	Sheathed and coppered.	Unsheathed.
Length on load water line, feet	502	502
Breadth, extreme, at load water line, about, feet	70	69–6
Trial displacement, about, tons	13,800	13,400
Mean draft at trial displacement, about, feet, inches	24–6	24–6

	Sheathed and coppered.	Unsheathed.
Greatest draft, full load, about, feet, inches	26–6	26–6
Total coal-bunker capacity, tons	2,000	2,000
Coal carried on trial, tons	900	900
Feed water carried on trial, tons	75	75
Speed not less than, knots	22	22

129. Summary of weight to be carried, Department's design:

	Tons.
Guns, mounts, shields, etc	407.5
Ammunition, ordnance stores, and outfit (two-thirds full supply)	288.0
Steam engineering complete, with water in boilers, condensers, piping, etc., and stores, etc., not to exceed	2,060.0
Reserve fresh water for steaming purposes (full supply)	150.0
Total coal	2,000.0
Boats and outfits	42.0
Masts and spars	24.29
Equipment, complete (including anchors, chains, electric plant, etc.)	305.83
Miscellaneous stores and water (two-thirds supply)	91.04
Provisions, clothing, and small stores (two-thirds supply)	122.42
Officers, crew and effects	89.65
Total protection, including armor, teak backing, armor bolts, cellulose, and splinter bulkheads	2,219.0

Dale.—The torpedo-boat destroyer *Dale* was successfully launched at 4·15 P. M., July 24, 1900, at the works of the Wm. R. Trigg, Co., Richmond, Va. Miss Mary Hasell Wilson, of Philadelphia, Pa., christened the vessel.

The *Dale* is 245 feet long, 23 feet 7¾ inches beam, and 6½ feet draft, and has a displacement of 420 tons. She is one of the sixteen destroyers contracted for in 1898, and will have a speed of 28 knots

St. Louis, Milwaukee and **Charleston.**—The new protected cruisers for the U. S. Navy, to be named the *St. Louis, Milwaukee* and *Charleston*, will be in general similar to the *Brooklyn*, but without the "tumble-home" or sloping sides of the latter. They will be of about 9,000 tons displacement, with engines of 20,000 horsepower for a 22 mile speed, or 23,000 horsepower for 23 miles. The bunkers will hold about 7,000 tons of coal. Light armor will protect the broadside guns, and there will be a heavy protective deck. The main battery will probably consist of six-

teen 6-inch rapid-fire guns, seven on each side, and one each at bow and stern. There will be no turrets, but all the guns will be protected by shields. It is possible that 8-inch or 10-inch guns may be placed at the bow and stern.

Kentucky.—In its report of the final trial of the *U. S. S. Kentucky*, at sea from Newport, R. I., June 25, the Board of Inspection and Survey, of which Rear Admiral Frederick Rodgers was president, says:

"The sea was smooth the entire trip and the weather cleared during the latter part of the first day. The winds were light and variable. At 2·30 P. M., June 25, the two hours' full power trial under natural draft began, with fires in all boilers, and a representative of the contractors in the engine room. The coal used was Pocahontas of average quality which had been lying in lighters for a long time and made much clinker and ashes. The machinery, both main and auxiliary, worked satisfactorily; some warming up of journals was probably due to inexperience of engineer's force.

"The average number of revolutions per minute of the engines for the two hours' run was 97.61—practically the same as made by the *Kearsarge* during her full power trial, 97.8. During the forenoon of June 26, the weather being clear, the firing tests were begun. It was decided to fire a salvo (all guns simultaneously) from each turret at 4,000 yards range, and then each gun singly at extreme elevation, beginning with the forward turret. When reported all ready in forward turret the command 'Fire!' was given from conning tower by 'battle order' system. The right 13-inch gun of forward turret fired, followed a few seconds later by the discharge of the right 8-inch gun of the same turret.

"The guns were loaded and given a common elevation. At the preparatory order 'Ready!' the breeches of the four guns were completely closed; whereupon, before the turret officer touched or closed the common ground button for firing, the right 13-inch gun discharged prematurely. Orders were given after a delay of twenty seconds to remove primers; when, in removing said primers, the right 8-inch gun also discharged prematurely."

After detailing the experiments undertaken to determine the cause of these premature firings, the Board says:

"Without changing other conditions, the polarity of one battery was reversed by reversing the battery terminal plugs, whereupon the primer of the corresponding gun fired, showing that it is necessary to insure the proper polarity of the batteries. The pole pieces of the cells should be therefore so marked or shaped as to avoid any mistake of misconnection. They are not thus now.

"After the 8-inch primer had been fired, the plugs in the battery, which had been reversed, were changed so as to bring its polarity back to that of the others, and, in so doing, one of the 8-inch guns was discharged. Further investigation appeared to show that during this operation one of the plugs touched a part of the enclosing case of the battery which was grounded and thus completed the circuit, causing the primer to fire. This shows that the battery should be carefully and completely insulated from the ship. It also shows that, with the present arrangement, a ground in any gun may fire another gun. The difficulty might be obviated if, instead of having similar poles, permanently connected together before passing through the firing key, they were each carried to the clips of a switch, which, when opened, would disconnect them, and when closed connect them together and then close the ground circuit for the common discharge.

"On the discharge of the right 13-inch gun at 4,000 yards elevation, the front cap square bolt of left trunnion was broken. These bolts were the original ones fitted to the mounts. Two (2) new bolts were fitted to this cap square and the guns made ready for firing. The test of the broadside guns were proceeded with, and each gun of the main and secondary battery was fired at level and at extreme elevation.

"In after turret this salvo was successfully fired, and the 13-inch guns fired at 10,000 yards and 8-inch at extreme elevation. No casualties occurred in this turret. All the projectiles of the 'salvo' fell in the same line and appeared to fall in the same spot. When the bolts were replaced in the forward turret, the

13-inch and 8-inch guns were fired at 4,000 yards and at extreme elevation. After this firing it was found that the rear left cap square bolt (a new one) of the right 13-inch gun (same gun) was broken."

Various defects in the vessel are noted as needing correction, and the Board says: "The same excessive jump to the mounts of the automatic one-pounders in tops was observed as in the case of the *Kearsarge*. The automatic 6-millimeter Colt guns in the top failed to work properly, only a few discharges taking place in each gun when a cartridge would jam, necessitating long delays in extracting jammed cartridges, and two (2) guns in forward would not permit belts to be entered at all. During the turret-gun test the blasts from the 8-inch guns did not inconvenience the persons at the 13-inch guns.

New Submarine Boats.—The Department has selected the following names for the six submarine torpedo boats, to be built by the Holland Torpedo Boat Company under authority of the Acts approved June 10, 1896, as amended March 3, 1899, and June 7, 1900, viz: No. 3, *Adder;* No. 4, *Grampus;* No. 5, *Moccasin;* No. 6, *Pike;* No. 7, *Porpoise;* No. 8, *Shark.*

U. S. S. Thornton.—At the Trigg yard, Richmond, Va., the U. S. torpedo boat *Thornton* was launched May 15. The ceremony of christening the new boat was performed by Miss Mary Thornton Davis, grand-niece of Capt. J. S. Thornton, U. S. N., who was executive officer of Admiral Farragut's flagship at the battle of Mobile Bay, and was mentioned in despatches. The *Thornton* is one of the group of boats authorized by act of Congress of May 4, 1898. A contract for her construction was signed November 16, 1898, and her keel was laid March 16, 1899. Her dimensions are: Length, 175 feet; beam, 17 feet 6 inches; mean draught, 4 feet 8 inches; displacement, 165 tons. She is to be fitted with twin screws and quadruple-expansion engines, and Thornycroft boilers. The estimated trial I.H.P. is 3,000 and the speed 26 knots.

Alabama.—The official full-power trial of this battleship took place on Tuesday, August 28, 1900, over the Cape Ann to Cape Porpoise course, and was a most satisfactory and successful test,

resulting in the accomplishment of a speed of about 17 knots, a knot over the contract speed, and this without distress of any kind to the machinery. A full report of the trial will be given in the November JOURNAL, it being impossible to secure the full data in time for the present issue.

Secretary Long recently announced that the President had selected the names for the eight war ships authorized by the last naval appropriation act, as follows:

Battle Ships.—No. 16, *Virginia;* No. 17, *Rhode Island.*

Armored Cruisers.—No. 7, *Maryland;* No. 8, *Colorado;* No. 9, *South Dakota.*

Protected Cruisers.—No. 20, *St. Louis;* No. 21, *Milwaukee;* No. 22, *Charleston.*

AUSTRIA.

Kaiser Karl VI.—The Austrian armored cruiser, *Kaiser Karl VI*, was placed in commission at Pola, May 28, 1900, after a successful series of progressive trials.

This vessel was launched at the yards of the "Stabilimento Tecnico," at Trieste, October 4, 1898, and was built by that firm from the plans of Chief Engineer Kellner. Her principal dimensions are: Length over all, 390 feet; length between perpendiculars, 367 feet 6 inches; beam, 56 feet 4 inches; draught (mean load line), 20 feet 5 inches; displacement, 6,325 tons.

The engines were designed by Director Landecke. The boilers were furnished by Maudslay Sons and Field. The hull is made from the best domestic Siemens-Martin steel and has several watertight compartments. Below the engine and fire rooms and the ammunition store rooms, for about 70 per cent. of the ship's length, the hull has a double bottom. In this double bottom, every fourth, and higher up, every second frame is made solid and watertight. By this subdivision into many watertight compartments, and by the large pumping facilities at command, the safety of the ship seems to be well assured.

To obtain a high degree of stability, the ship is provided with bilge keels of 35.4 inches height for about half the length of the vessel. On both sides of the hull and for about 70 per cent. of its length there is an armored belt of nickel-steel "Witkowitz"

of 78.6 inches height and 8.65 inches maximum thickness. It extends to a depth of 45.2 inches below the water line and ends forward and aft in an armored transverse bulkhead from 6.88 inches to 7.86 inches thick. The armored belt is joined at the top to a central protective deck 1.257 inches thick, that ends forward and aft of the transverse bulkheads in a protective deck of 2.34 inches thickness. Above the belted armor, on the middle deck, there is a casemate 3.13 inches thick on the outside and 2.34 inches thick on the forward and aft transverse bulkheads. The four sponsons of the battery deck, which each carry 2 rapid-fire guns of 5.9 inches are provided with an armor 3.15 inches thick. The several gun mounts are separated by special splinter shields .788 inch thick. The two Barbette turrets have an armor of 7.88 inches and carry each one 9.45-inches rapid-fire gun L/40. Their substructure is protected by an armor 4.72 inches thick. The ammunition passages of the 5.91-inch guns are provided above the middle deck with an armor 1.18 inches thick; the hatchway coamings for the entrances to the engine and boiler rooms have armor plates 3.94 inches thick. The weight of the entire armor is 23 per cent. of the displacement, or over 1,400 tons.

The stem and sternpost, the rudder and the out-board shaft supports are made of soft steel castings of domestic manufacture; the shaft tubes are made of welded wrought-iron tubes. The rudder has a surface of 200 square feet. Two steering engines are provided to give additional security of action. Electrical machinery is installed for driving the ventilating fans, for the ammunition hoists of the heavy guns, and for their handling.

The coal bunkers have a capacity of 820 tons. The ship carries one signal and one military mast; in the fighting top of the latter there are two 1.85-inch quick-fire guns and two .315-inch machine guns, "system Skoda."

The main battery of the cruiser consists of two Krupp 9.45-inch rapid-fire guns L/40 using fixed ammunition. They are installed in fixed turrets and fire "en barbette" with an angle of action of 130 degrees from the keel line forward and aft respectively.

The initial velocity of 474 pounds projectile is 2,300 feet, by which a total energy of 15,760 foot tons, and a penetration of 34.6 inches wrought-iron is obtained. The greatest elevation is 20 degrees, and the range at this elevation is 46,000 feet. The turning of the turret roofs, the elevation of the guns, and the hoisting of the ammunition is effected by secondary dynamos, that are connected to primary dynamos installed below the protective deck. A secondary battery consists of eight 5.91-inch rapid-fire guns L/ 40, "system Skoda," so arranged that four of them together can fire in a direction forward or aft or athwart ships. Each of these guns is able to fire at least six aimed rounds per minute. Two 2.76-inch guns serve as boat and landing armament; eighteen 1.89-inch rapid-fire guns and two machine guns complete the armament. The torpedo outfit consists of two broadside launching torpedoes.

The electric-lighting system contains about 350 incandescent lights; the four searchlights have arc lamps of 25,000 candlepower each, with projectors of 23.7 inches diameter. The power is provided by two dynamos of each 45,000 volt-ampères. In addition, the four dynamos that serve the 9.45-inch guns can be added to these in emergencies. A night-signal apparatus, system "Sellner," is included. In the steam launches of the ship there are also provided small dynamos supplying searchlights of 4,000 candlepower with 13.8-inch projectors.

The machinery consists of two vertical, four-cylinder, triple-expansion engines, driving two screws. The cylinders are 35.82 inches, 55.52 inches, 60.63 inches, and 60.63 inches diameter, and have 37.4 inches stroke.

There are three fire rooms, containing eighteen Belleville water-tube boilers, with economizers; there are fifty-four furnaces; the total heating surface is 35,000 square feet, and the total grate surface is 918 square feet; the ratio of the H.S. to G.S. is 38.15 to 1.

The steam pressure in the boilers is 260 pounds per square inch and is reduced at the engines to 177.6 pounds per square inch. The boilers contain 4,982 water tubes of a total length of

34,750 feet. The two 3-bladed propellers are of 16.01 feet diameter and have a pitch adjustable from 15.54 to 19.15 feet.

The total weight of the steam machinery is 1,110 tons. In addition to the main engines there are 70 independent auxiliaries having in all 126 steam cylinders.

The official contract trial took place at Pola April 12 and 21, 1900. The progressive trial was had April 12 with a light sea and a S.E. wind of one-half force. The draught was 18.94 feet forward, 22.26 feet aft, corresponding to a displacement of 6,325 tons. The results of this trial are contained in the following table:

No. of run.	Revolutions per minute.		Indicated horsepower.			Time over 3-knot course.						Average speed in knots per hour.	
						1.		2.		3.			
						Knot.							
	port.	*stbd.*	*port.*	*stbd.*	*total.*	*min.*	*sec.*	*min.*	*sec.*	*min.*	*sec.*	*avge.*	*mean,*
I	61	60	694	555	1,249	5	46	10	30	17	8	10.51	10.16
	61	60	709	529	1,238	6	21	12	23	18	20	9.82	
II	80	80	1,388	1,343	2,731	4	19	8	39	12	56	13.92	13.52
	81	82	1,425	1,388	2,813	4	37	9	13	13	43	13.12	
III	102	100	3,039	3,082	6,121	3	34	7	1	10	27	17.22	16.66
	101	102	3,020	2,502	5,522	3	54	7	34	11	11	16.10	
IV	112	108	3,975	3,477	7,452	3	19	6	23	9	42	18.56	18.17
	111	110	3,982	3,464	7,446	3	25	6	46	10	7	17.79	
V	120	123	5,012	5,448	10,460	3	0	5	59	8	58	20.07	19.71
	121	120	5,047	4,999	10,046	3	7	6	13	9	18	19.36	
VI	126	126	5,865	5,879	11,744	2	52	5	44	8	37	20.89	20.56
	126	128	5,527	5,861	11,388	3	1	5	58	8	54	20.23	
						← knot.		*min.*	*sec.*				
VII	21	20	73	65	138			0	10			3.27	3.62
	19	20	56	60	116			7	32			3.98	

April 21 followed the four hours' endurance trial, which took place over a 17-knot course from Olmo to Guistiguan and which was marked by three tenders. The ship had a draught of 18.94 feet forward, 22.26 feet aft, corresponding to a displacement of 6,325 tons. The sea was light, the wind E., of 3–5 force, and the current about one mile strong.

Kaiser Karl VI left the central harbor at 7·00 A. M., and entered upon the 17-knot course with full power at 9·00 A. M. The direction of the course was N.W. 47 minutes 46 seconds, returning at 50 minutes 29 seconds.

Revolutions per minute, average,	129.44
Indicated horsepower, average,	12,900
Mean boiler pressure, pounds per square inch,	219
Vacuum, inches of mercury,	27.2
Greatest speed in knots,	20.826
Air pressure, inches of water,	3.94

With the exception of a few minor leaks at the junction boxes of the boiler tubes, boilers and engines worked to perfection. The management of the machinery was in the hands of the employees of "Stabilimento Tecnico Triestino," under supervision of the Imperial Navy. The firing of the boilers was done by a force, per watch, of 24 firemen from the navy and 6 firemen furnished by the contractors; the coal supply was handled by 12 trimmers from the navy and 6 men from the contractors.

The coal was Nixon's double sorted of latest importation. Temperatures were taken during the trip in several localities of the ship and at no place were they found to be excessive.

The results of both trial trips were pronounced as eminently satisfactory by the committee on acceptance.

The cruiser *Kaiser Karl VI* is a valuable addition to the Austrian Imperial Navy, for she has a high fighting value, due to her very heavy armament, her strong armor, her high speed and her great steaming radius.

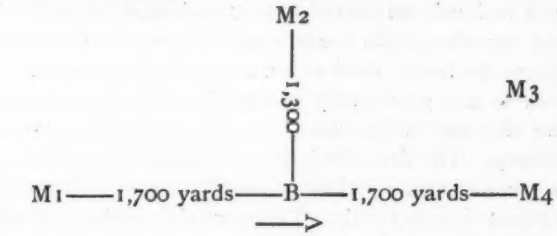
ENGLAND.

The Belleisle Experiment.—An experiment of singular magnitude and importance was made on May, 26, 1900, by the British Admiralty, in the form of a direct attack upon one of the older English battleships, the *Belleisle*, by the first-class battleship *Majestic*, the former being moored in shallow water off Selsey Point, near Portmouth dockyard, and the attacking vessel steaming by at a 6-knot speed, with a range of from 1,700 to 1,300 yards.

In recapitulating the data of this experiment the "Engineer," of London, of June 8, gives the following terse account and comment.

The vessel was subject to nine minutes' consecutive firing, instead of the series of two minutes' fire that formed the original programme. Why the change was made is not quite clear; it is generally regretted, as there was rather too much "killing the dead horse" about it for at least half the period. That is to say —saving for one hit—practically everything of moment was accomplished in the first two or three minutes.

The following diagram represents the course and ranges:



The *Majestic* steaming firing from M1 to M3, the *Belleisle* (B) being moored with her bow in the direction of the arrow. All the serious hits appear to have taken place between M1 and M3. The *Majestic* steamed at six knots. So far as can be ascertained the *Majestic* fired:

8 rounds	12-inch common shell.
7 rounds	12-inch A. P. shot.
About 100 rounds	6-inch lyddite (at bow and battery).
About 100 rounds	6-inch common (at stern).
400 rounds	3-inch various.
750 rounds	3-pounder shell.

And an indefinite number of Maxims. No torpedoes were discharged. Full charges were used throughout for all guns.

Roughly speaking, about 30 to 40 per cent. of the projectiles fired were effective, the rest went over or just short, or else went clean through and burst beyond. Only about 10 per cent. of the projectiles were clear misses outside the black and white haystack of smoke that shrouded the target.

Here, perhaps, we should recapitulate the data of the *Belleisle*. She is of about 5,000 tons displacement, and the iron armor is

disposed as follows: Belt, 12 inches, 6 inches; battery 10 inches, 8 inches; redoubt below battery, 9 inches, 8 inches; conning tower, 12 inches. The belt rises about 2 feet to 2½ feet above the water line, but at the time of the trial about 4 feet were shown. The redoubt is octagonal like the battery. The conning tower is situated under the bridge on top of the battery; it is very low, and the hood stood at least 18 inches above the walls.

Taking the 12-inch hits first, the demolition of this erection in the bow is variously attributed to a 12-inch and a 6-inch. We think, however, that it was due to one of the latter. On the belt, below where the funnel used to be, two shells have burst. One or both these are presumably 13-inch hits, there being in one case—the after one of the two—a 12-inch penetration of about 2 inches deep. The armor belt just below is cracked laterally for some 7 feet, and a vertical seam has also opened. The lateral crack is about 6 inches wide. There are the marks of a 6-inch shell in the aperture. The other hit has no mark beyond the abrasion caused by the burst.

On the water line, directly under the battery, there is a large hole made by a 12-inch shot by which the ship eventually sank; though not, as erroneously supposed, the direct cause of her sinking. The second 12-inch projectile fired blew the side right in, sending one fragment of armor right up through the flat armored deck above it. It did not, however, send anything down into the engine room. The belt here offered a resistance equal to from 9-inch to 12-inch Harvey, because it was struck at a considerable angle—one, indeed, in which we might conceivably have expected the shot to glance off.

Immediately abaft the casemate a 12-inch shell has blown away some 10 feet square of side, which here looks something like the ruins of a recessed port.

Well aft, another 12-inch shell has hit. Here, too, a large piece of side is blown in, the deck above has disappeared, and the iron deck beams are hurled up into shapeless remains more like shavings than anything else. The external area of damage is about 10 feet; internally the whole of the cabins hereabout were reduced to match wood.

There remain two more hits that may be 12-inch. At the after end of the unarmored top of the battery, and towards the after end of what little remains of the funnel base, there are huge irregular holes, averaging 5 feet to 7 feet each. Most of the edges are bent inwards.

We now come to the most mysterious hit of any. High up on the port side after porthole is a clean, nearly exactly circular penetration of exactly 10 inches in diameter. There is around it an irregular depression, with a maximum depth of 2½ inches, and some 2 feet across at the widest. What made this hole no one seems inclined to be positive about.

On the upper and water edge of the after part, on the starboard side of the battery, another projectile has chipped out a piece of armor and burst, judging by the marks. This is the total of the big projectiles, three of them certain, and five of varying degrees of depth.

The bow and battery were fired at by 6-inch lyddite, the 6-inch common shell were directed aft. Between the damage done by one and the other there is no comparison. Both have made holes in the armored ends; both have done no harm at all to armor, both have made matchwood of the woodwork inside. But while the 6-inch common has destroyed as one might destroy a wooden box with an axe, lyddite has pulverized completely. Practically there are no remains beyond a dust. The lyddite has acted in a moment much as dry rot acts in a score of years.

It is noteworthy that there is not a single sign of charring. It is not merely that no burnt wood is visible, there are no signs of scorching either, though possibly a week's bleaching under the sun may account to some extent for this.

Where 6-inch common shell have burst between decks, the deck above shows no sign of it, but it is quite another story with lyddite. Not only are huge holes blown upwards, but the entire deck is bulged up. The cross beams were thick and plentiful, but their resisting power has been nil.

The extreme bow is absolutely untouched. Abreast of the funnel—curiously enough just where the armor is thickest—

there are several dents where shells have gone in an inch or two. The masts are in a bad way. The foremast is the least damaged. The yards on both masts were shot away, and both peaks were dropped. The mainmast is especially in a very bad way, and had to be shored up. It was cut in two, all but a tiny piece, low down, and remained standing only because all the shrouds were not destroyed.

The only boat left is the steam pinnace carried between the battery and mainmast. A 6-inch common shell entered this boat and blew the starboard side of it into a number of shapeless streamers. It is absolutely clear that no ordinary boats will float after an action; it is doubtful if they will exist.

The damage to the funnel cannot well be described, as there is nothing there but an enormous hole, with strips of iron twisted over it. The bridge, though much bent and twisted, survived, but the searchlight has blown away.

The upper works were blown to pieces, the 6-pounders either knocked clean away or else blown sideways with some of their fittings melted. Ventilators and so forth were pitted with innumerable small holes from splinters and fragments.

Inside the battery, in which at least one shell that had entered a port hole had burst, all the gun sights were destroyed. The guns themselves are absolutely unhurt, but, owing to damages to sights and gear, would have been of no use at all. The dummies around them were all burned.

It is worth noting that two davits have been cut clean in two by projectiles. A great many small projectiles embedded themselves in places, and a few still so remain. Most, however, were shaken out.

Directly firing ceased the ship was boarded by fire parties and by the officers told off to inspect her. She was full of dense smoke, caused by the exploded shell, by the pierced funnel base, by a few small bonfires of dummies and a lyddite shell or two that had failed to burst properly and were burning. Practically nothing was to be seen for the smoke and fume.

It was taken for granted that the ship was on fire, and water was pumped into her. This lowered her freeboard, and the sea

then rushed in at the hole on the water line made by the 12-inch shot.

The ship sank just after the last of the exploring party had left her—sunk, not by gun fire in the picturesque fashion that the daily papers described, but in a prosaic attempt to extinguish fire.

Some of the *Belleisle's* hoses were still working when the ship was boarded, others were found cut to pieces. Most of the valves and so forth had disappeared.

A side experiment in connection with the *Belleisle* was to test the efficacy of the Douglas leak stopper, which works more or less on the system of an umbrella. Only two were actually made use of. These were fitted in side holes just before the funnel, and answered with perfect success. It is unfortunate, however, that a far more ample test was not made.

The report about Krupp armor being placed on the battery is pure fiction. The torpedoes were actually in the tubes, and they were also fully loaded except for the pistols. One was found still in the tube, the other was gone.

The *Belleisle* experiment has been a triumphant vindication of certain British systems of ship construction. The *Bouvet*, *Charlemagne* or *Kaiser Friedrich der Dritte*, had any of them been the target, would have fared as badly—very possibly worse than the *Belleisle;* and there is no reason why that should be concealed. The British, Japanese, United States, and to some extent the Russian and Italian are the only navies that count—from the constructional standpoint—in the light of the *Belleisle* experiments.

It is understood that further experiments are to be made on the *Belleisle*. The next trials will be with torpedoes, the object apparently being to ascertain the effect upon a warship if a torpedo exploded before being discharged from the tube. Torpedoes fitted with warheads are to be placed in the *Belleisle's* tubes, and these will be fired at by small guns in order to explode the gun-cotton charges in the heads.

Aboukir.—On the 16th of May, 1900, the first-class armored cruiser *Aboukir* was launched from the yard of the Fairfield Ship-

building Company at Glasgow. She is a sister ship of the *Cressy*, which was launched from the same yard on the 4th December, last year, and the following description will answer for all the vessels of the class:

The *Aboukir* is one of a class of six similar vessels at present under construction in different yards throughout the country, and is of an entirely new type. Her length between perpendiculars is 440 feet; beam, 69 feet 6 inches, and draught, 26 feet 3 inches, with a displacement of 12,000 tons, of which 7,860 tons will be absorbed by weight of hull alone.

The main armament will consist of two 9.2-inch breech-loading guns, one firing ahead and the other astern, both being mounted in 6-inch armored barbettes, with shields. These weapons fire a projectile weighing 380 pounds, with a muzzle energy of 14,520 foot tons. Four 6-inch quick-firing guns will be fitted in 6-inch casemates to fire direct ahead, and four to fire astern, each gun being capable of training to fire over a large arc on the broadside. In addition to these there will be four similar guns amidships, two on either side, besides twelve 12-pounders and a large equipment of machine guns. Protection will be afforded by an armor water-line belt 6 inches in thickness, of Harveyized steel, manufactured on the principle introduced by Mr. Willlam Beardmore, of Glasgow, terminating 120 feet from the bow and 90 feet from the stern in transverse bulkheads of 5-inch steel, the depth of the belt extending to 5 feet below the normal load water line, and 6 feet 6 inches above the load line, the whole length of the belt being 230 feet. All four sides of the oblong fighting citadel are thus armored. The armor is backed with 4 inches of teak, so that with the wood sheathing on the outside there is a total thickness of 15 inches. In addition to the side armor, the bow is plated with 2 inch nickel steel, specially hardened, which will even withstand attack from a 6-inch quick-firing gun. This nickel-steel plating is 14 feet 6 inches deep at the transverse bulkhead, 120 feet from the bow, and widens out so as to be full depth of the ship at the ram. The protective deck, which is curved downwards at each side to the bottom of the broadside armor, is 1½ inches within the fight-

ing citadel, where the 6-inch side armor forms the chief protection; forward and abaft this it is from 1½ inches to 3 inches thick, and in the center of the ship the crown of the arch is one foot above the water line. The main deck above this is also specially strengthened.

The ship will be engined by the builders with two sets of triple-expansion machinery, to develop, collectively, 21,000 I.H.P., the resulting speed being estimated at 21 knots. The maximum revolutions of the engines will be 160, and the working steam pressure 250 pounds per square inch. The boilers will be of the Belleville type, thirty in number, and the four compartments into which the boiler space is divided occupy 130 feet of the length of the ship. Provision is made so that the steam, when it has done its work in the auxiliary engines, will be passed into the evaporators and distillers before going to the condenser, and will thus do the further duty of making fresh water taken from the sea. There will be accommodation in the bunkers for 1,600 tons of coal.—"Naval and Military Record."

Hogue.—Her Majesty's cruiser *Hogue*, launched by Lady Muncaster on August 13, is a warship of the *Cressy* type. She is one of the first armored cruisers built since the *Australia* class was ordered over ten years ago, and constitutes an entirely new type of fighting ship. In many respects the *Hogue* resembles the *Powerful*, also built by Messrs. Vickers, Sons, and Maxim, Limited, embodying the best features of that magnificent ship, with the addition of an armored belt of considerable area. The displacement is 12,000 tons. The hull is sheathed with teak and coppered. The armament comprises 9.2-inch (22-ton) guns, each mounted in armored barbettes, the mountings being a special design of the Vickers Company, by which the guns can be loaded at any angle of elevation for training. These 9.2-inch guns fire a 380-pound projectile with a muzzle energy of 14,520 foot tons. There are also eight 6-inch guns, with a great range of fire, and there are distributed throughout the ship twelve 12-pounder quick-firing guns and a number of machine guns. The four boiler compartments of the *Hogue* take up 130 feet of the length of the ship, the coal bunkers being arranged on either side of

the boiler rooms, and over the protective deck; and an ammunition passage is situated immediately under the protective deck. There is also an athwart-ship bunker right forward. Thirty boilers are carried, all of the Belleville type. The boilers have been designed with the most liberal steam-generating surfaces, so that no difficulty should be experienced in obtaining the full power, and even more, if of any utility; and at the same time a higher power is obtained per ton of machinery than could be realized with ordinary boilers. At full power the engines will make 120 revolutions, which is estimated to drive the ship at 21 knots. The gun mountings, under manufacture by Messrs. Vickers, will be placed on board previous to the official trials being carried out, and when ready to leave Barrow, the *Hogue* will be delivered to Her Majesty's Dockyard, Devonport.

Falcon-Ostrich.—After a couple of preliminary runs the two torpedo-boat destroyers *Falcon* and *Ostrich*, built by the Fairfield Shipbuilding and Engineering Company, Limited, to the order of the British Admiralty, have during the past fortnight most satisfactorily passed through the official trials in the Firth of Clyde. The mean results were: *Falcon*, speed trial, 30.135 knots; coal-consumption trial, 30.09 knots. *Ostrich*, speed trial, 30.113 knots; coal-consumption trial, 30.161 knots. A special low-power coal-consumption trial of the *Ostrich* was also carried out at a speed of 13 knots, when the distance run per ton of coal was nearly 44 knots, equal to about 50 miles.

FRANCE.

Gloire.—The first-armored cruiser *Gloire* was launched at Lorient on the 27th of June. Her dimensions are as follows: Length, 452 feet 9 inches; beam, 65 feet 4 inches; displacement 10,000 tons, with a draught of 24 feet 6 inches. The engines, driving three screws, are to develop 20,000 I.H.P., giving a speed of 21 knots, her coal supply of 1,590 tons giving a radius of action of 10,400 miles at 10 knots, and 1,940 knots at full speed. Protection is afforded by a complete water-line belt of 5.9-inch steel, with an upper belt and bulkheads of 3-inch steel. Six-inch armor on the main turrets, and 4-inch on the casemates and sec-

ondary turrets. A 2-inch armored deck, with a 1.4-inch splinter deck. Her armament comprises two 194-millimeter (7.6-inch) in turrets, one forward and one aft; eight 164.7-millimeter (6.4-inch), two in casemates with recessed ports on main deck aft; two forward, and four in turrets on upper deck; six 100-millimeter (3.9-inch), eighteen 47-millimeter (1.8-inch), and six 37-millimeter (1.4-inch) guns, all Q.F. except the two 7.6-inch; and she has four torpedo-discharges, two of which are submerged. Her cost is 22,561,519 francs ($4,386,000).

Steam Trials.—The first class battleship *Hoche* has made a successful preliminary trial of her water-tube boilers, which are of the Belleville type. She is the first of the older battleships to have water-tube boilers substituted for her cylindrical ones.

The new destroyer *Fauconneau* has made several successful trials, maintaining a mean speed of 27.5 knots.

The new first-class gun boat *Zèlée* has concluded her trials; at one-third power the engines developed 335 I.H.P., with a coal-consumption per H.P. per hour of 574 grammes; during a twenty-four hours' run, the engines developing 550 I.H.P., the coal consumption per H.P. per hour was 727 grammes. During the six hours' full-speed trial the engines developed 953 I.H.P., giving a speed of 13.5 knots with a coal consumption of 721 grammes per H.P. per hour. Her boilers are of the Niclausse type.

Her sister ship, the *Dècidée*, has also successfully completed her trials. She also has Niclausse boilers, but the coal consumption has somewhat varied. At one-third power the consumption was 687 grammes per H.P. per hour; at 500 I.H.P., the consumption was 767 grammes per H.P. per hour, and at the full-speed trial it rose to 818 grammes per H.P. per hour, the speed of the ship being 13 knots.

Jules Ferry and Léon Gambetta.—Orders have been issued to Cherbourg and Brest to lay down and commence the two first-class cruisers *Jules Ferry* and *Léon Gambetta*. These two vessels are sisters and identical in all respects. They will have a displacement of 12,416 tons, their length will be 475 feet, and beam 69 feet, 24,000-H.P., estimated speed, 21 knots. They will carry

1,950 tons of coal, giving a radius of action at 10 knots of 10,000 miles, and at full speed of 1,325 miles. Their armament will comprise four 7.6-inch, sixteen 6.4-inch, twenty 1.8-inch, and four 1.4-inch guns, and they will have five torpedo discharges. The complement will be 23 officers and 687 men, and the cost of each amounts to 28,982,500 francs. The *Ferry* is to be completed at Cherbourg early in 1905, and the *Gambetta* at Brest by the middle of 1904.

GERMANY.

Wittelsbach.—The German battleship *Wittelsbach*, hitherto known as *C*, which was launched at Wilhelmshaven on July 3, is the first of a new type of battleship. She is to be followed by four others of the same type, which are being built respectively at Dantzig, by Schichau; at the Germania yards, Kiel; at the Vulcan yards, Stettin; and at the Imperial Government yards, Wilhelmshaven, where the *Wittelsbach* was built. The *Wittelsbach* has 800 tons greater displacement, and her engines have 2,000 horsepower more than the ships of the *Kaiser* type, some of which are still building. There is also a considerable difference in her armor and armament. Her length is 418⅔ feet,; beam, 68⅔ feet; draught, 19 feet, and displacement, 11,800 tons. She has had 5,480 tons of steel worked into her. Her armor deck is from 1.6 inches to 3 inches thick. Her engines develop 15,000 horsepower, giving her a speed of 19 knots, which is one knot more than the *Kaiser* class. She can carry 1,000 tons of coal if necessary, 650 tons being her normal load. Her armament will consist of four 9.6-inch, eighteen 6-inch, twelve 3.52-inch, twelve 1.48-inch, and eight machine guns. The four 9.6-inch guns will be placed in two turrets, with a 10-inch protection, the guns in the fore turret being nearly 30 feet above the water, and those aft 21½ feet. She will carry six torpedo tubes, one forward and four broadside submerged, and one aft above water. Her complement will be 65 officers and 650 men. Her cost is £1,118,000, of which £832,500 is for the ship, and £35,500 for the torpedo armament. She will be complete with all her armament by April 1, 1902.—"Engineering."

Cost of Construction.—The first of the four battleships

under the German navy-extension scheme has been ordered at an average cost of $5,600,000. The Vulcan Company, of Stettin, Schichau, of Danzig, and Krupp, of Germania, having secured an order for one each; while the fourth is to be built at one of the Imperial dockyards. The price works out to nearly $475, while the latest British ships, complete, cost $378 per ton displacement. British ships of 15,000 tons displacement cost little more than the German ships of 11,800 tons. The German ships have a more numerous, although not more powerful, armament; but the cost of guns in the one, with mountings, etc., is $1,250,000, and for the British ships $630,000, or about half. The British ships carry four 12-inch guns, the German vessels four 9.6-inch guns; but many have eighteen quick-firing guns to twelve for the British ships. The smaller guns were practically alike in both types—twelve of about 3-inch bore and twelve smaller. The Germans have six torpedo tubes to four in the English ships, all submerged. The speed in both types is the same—19 knots, and in both cases the engines develop 1¼ horsepower for every ton to be driven at this speed.

RUSSIA.

Bayan.—The Russian first-class cruiser *Bayan* was launched at La Seyne on June 12, 1900. The engines are supplied with steam by twenty-six Belleville boilers and develop 16,500 horsepower, with a speed of 21 knots. The armor belt is 6 feet wide, 4 feet being below the water and 2 feet above, and varies from 4 inches to 8 inches in thickness, and above is a slighter protection, 3.2 inches thick. The deck is from 1.2 inches to 2 inches in thickness, and the protection of the turrets is 6 inches. The guns are loaded and the turrets turned by electricity. There are six searchlights.

Variag.—Unfortunately, an accident to the port H. P. cylinder of this new cruiser during her official trial, July 28, interrupted the most remarkable speed record of any modern cruiser. For more than seven hours and a half she had maintained a speed of considerably over 23½ knots, and every indication up to the time of the accident pointed to an easy accomplishment of the

requirements for a 12-hour continuous run at 23 knots, and the probability of reaching a much higher average speed.

Much interest centers in the results of the trials of this vessel, it being the first cruiser on which the performance of the Niclausse boiler can be observed by our own people. The JOURNAL expects to publish in the next issue a complete report of the official trial, which probably will soon take place, and will reserve data of the vessel until that time, with the exception of the following principal dimensions:

Length over all, feet and inches	425–1
L.W.L., feet and inches	419–5
Draught, mean, feet and inches	19–6
Displacement, tons	6,500
Engines	twin-screw, triple-expansion type.
Cylinders, diameter, H.P., inches	40
I.P., inches	62
L.P. (2), inches	68
Stroke, inches	36
Normal I.H.P.	20,000
speed, knots	23

There are thirty Niclausse boilers with a total heating surface of 622,290 square feet and a grate surface of 1,575 square feet. Steam pressure is 250 pounds per square inch. Normal coal supply, 700 tons.

MERCHANT STEAMERS.

The Burning of the North German Lloyd Steamships and Docks, Hoboken, N. J.—The record of disasters from fire in New York contains nothing more terrible than the conflagration which occurred on the afternoon of Saturday, June 30, 1900, by which between two hundred and three hundred lives were lost, many of them amid circumstances of unspeakable horror, while three of the largest steamships of the North German Lloyd Company, and the four piers which constitute the landing facilities of the company at this port, were destroyed.

The docks of this great company, which, together with those of its rival, the Hamburg-American line, are located on the New Jersey shore of the North River, at Hoboken, consisted of four large piers carried upon piling. The largest of these was a new, double-deck structure, only recently completed, while the others had been more or less renovated, and in one case considerably lengthened. Most of them were of the standard type of construction, which, unfortunately, in the light of the recent disaster, is used almost exclusively on the water front of New York harbor. Piling is driven over the whole area to be covered by the piers and capped with 12 by 12 timbers, upon which are laid the joists and wooden flooring. The sheds on the majority of the older piers are built with wooden posts and roof trusses, and are wood sheathed.

At the time of the outbreak of the fire, which, according to eye-witnesses, started about five minutes to four on the afternoon of June 30, the situation at these docks was as follows: To the north was the new pier, about 800 feet in length, recently completed by the company, and generally known as the Thingvalla pier. On the south side of it was lying the steamship *Main*, a new vessel of 6,398 tons, recently constructed by Blohm & Voss, of Hamburg, which came to New York on her

maiden trip only last April, and on this fatal afternoon was loading for her second return voyage across the Atlantic. She had accommodations for 300 cabin and 800 steerage passengers, and was in every respect a first-class ship. Next to the steamship *Main* was pier No. 1, with the steamer *Bremen* lying to the north of it, and the celebrated *Kaiser Wilhelm der Grosse* moored on its south side. The *Bremen*, built in 1896 by Schichau in Germany, was one of four huge vessels constructed to carry a large quantity of cargo and a considerable number of passengers. She was 500 feet in length, 60 feet in beam, and had a register of 10,000 tons, and a speed of 15 knots. The *Kaiser Wilhelm der Grosse* is to-day the fastest of the great ocean liners, and one of the most popular passenger vessels afloat. She is 648 feet in length, 66 feet in beam, and has a displacement of 20,000 tons. With a horsepower of 27,000, she has a record speed across the Atlantic of over 22½ knots an hour.

The next pier to the south, No. 2, was accommodating the steamer *Saale*, of 5,267 tons register, one of the older vessels of the company, which was constructed in 1886 at Glasgow. She is 455 feet in length and 48 feet in beam. Only a few hours before the disaster the steamer *Aller* had sailed from her berth between piers Nos. 2 and 3. At pier No. 3 there were several lighters and barges. It should be mentioned that all of these steamers had coal barges and canal boats alongside, from which they were taking coal and cargo, in preparation for the return trip across the ocean.

It seems that on the fatal afternoon there was lying in the middle of pier No. 3, a large quantity of cotton bales, and it is said that not far from this were over a hundred barrels of whiskey. It is generally agreed that the fire originated in the cotton, and it is supposed that the extraordinary rapidity with which it spread was due to the explosion of the whiskey and scattering of the inflammable contents over the cotton. Be this as it may, it is certain that the fire swept through the whole length of the pier with extraordinary rapidity. It is stated by a captain of a tug boat, that, although he was only a quarter of a mile from the pier when the smoke first appeared, and made all

haste across the river, the structure was wrapped in fire from end to end by the time he reached it. The spread of the fire was greatly accelerated by a strong westerly wind, and it is stated by eye-witnesses that within nine minutes from the time the first blaze was seen, the whole four piers and the steamships adjoining were ablaze. So swiftly did the flames leap from pier to pier and to the adjoining ships, that within a few minutes the *Main*, *Bremen*, *Saale*, and even the great *Kaiser Wilhelm der Grosse*, were afire.

On account of her extreme value, the efforts of the tugboats that rushed to give assistance were directed principally to saving the *Kaiser Wilhelm der Grosse*, and, although she was dragged out into the midstream with her upper works ablaze, the fire was got under control before very serious damage was done. Not so, however, with the other vessels. Although the moorings had been immediately cast adrift by the crew when the fire started, the concentration of effort on the *Kaiser Wilhelm der Grosse* and the efforts of the tugs to save the people who had leaped from the burning piers into the water, resulted in these three vessels being left to the mercy of the fire, until, under the action of the tide, two of them, the *Bremen* and the *Saale*, drifted out, blazing from stem to stern, into the Hudson River. The *Main* was burned at her dock.

While the loss of property was enormous, the magnitude of the disaster is to be measured by the awful loss of life and the aggravated horrors that attended it. Scattered over the ships, the piers and the lighters were the crews and longshoremen, busy at the work of loading; and the spread of the fire was so sudden that escape from the piers to the shore was impossible. As a consequence, the majority of those who were cut off from escape by land leaped into the water, many of them being drowned before they could be picked up; but even this relatively merciful form of death was denied to many unfortunates who were working in the holds and lower decks of the vessels, and only discovered what was going on when they found their escape through the hatches cut off by a sea of fire which was raging in the upper works. Many of these were doubtless suffocated by smoke or

burned as the vessels sank under the streams of water which were being poured into them from the fireboats and tugs. A feature which rendered the disaster particularly horrible was the fact that many of the imprisoned people could have been rescued had the portholes of the *Saale* been a few inches larger than they were. As it was, many victims died before the eyes and within actual touch of would-be rescuers, who were quite powerless to help them.

Within a couple of hours the whole four docks of the North German Lloyd Company and their warehouses were completely destroyed, and three of the passenger steamers were so far burned that it is as yet questionable, in the case of some of them, whether they will be worth refitting. The total loss of the company, including cargoes, is estimated by the general agent of the line to be about $5,000,000, while two hundred and fifteen souls perished —"Scientific American."

S. S. Vaderland.—The twin-screw steamer *Vaderland*, which was launched July 12, 1900, by Messrs. John Brown & Co., Limited, Clydebank, for the International Navigation Company's Red Star service between New York and Antwerp, belongs to a type of transatlantic liner which is greatly increasing in favor, combining, as they do, a satisfactory speed and magnificent accommodation, with considerable cargo capacity. The *Vaderland* can take 11,000 tons of cargo, as well as 342 first, 194 second and 626 third-class passengers, and her speed will enable her to leave Antwerp and New York, respectively, on Saturdays, and always land her passengers on Monday mornings; so that if Sunday is looked upon as a *dies non*, the journey will only occupy one week, about the time taken by the fliers if their time is similarly reckoned between their terminal ports on both sides.

The dimensions of the new vessel are as follows: Length between perpendiculars, 560 feet; breadth, 60 feet; depth, 42 feet. The gross tonnage is about 12,000.

Propelling Engines.—The main propelling engines are of quadruple-expansion type, direct-acting and surface-condensing. The four cylinders each work a separate crank, and they are so arranged as to produce a minimum of vibration, the parts being

so disposed as to be practically balanced without the use of counterweights on the crank arms. The order of the cylinders from the forward end is: High-pressure, second intermediate-pressure, first intermediate-pressure, low-pressure. The sequence of turning is high-pressure, low-pressure, second-intermediate, first-intermediate. The cranks are at right angles to each other. The diameters of cylinders are as follows: High-pressure, 31 inches; first intermediate, 44 inches; second intermediate, 62 inches; and low-pressure, 88 inches, with a stroke of 4 feet 6 inches. The high-pressure and the two intermediate-pressure cylinders are each fitted with piston valves, and the low-pressure has two triple-ported flat valves. All the valves are worked by a radial valve gear, operated by a single eccentric through a quadrant rocking on trunnions. The reversing is obtained by moving the sliding block attached to the valve spindle from one end of this quadrant to the other. The lap and lead are obtained by a separate lever worked from crosshead of the main engine. This arrangement of valve motion allows of the valves of the engine being placed at the back, thereby bringing the centers of the cylinders very close together, and economicing space in the engine compartment, and, of course, adding to the cargo capacity in the ship. The total length of the engines is about 29 feet, which is much less than could be obtained if the ordinary link gear had been used. The reversing gear is controlled by a steam direct-acting engine.

The shafting is of Siemens-Martin steel. The four cranks are each built up separately, and are interchangeable. The crank shaft is 17¼ inches in diameter, the thrust shaft is 17¼ inches, and the tunnel shaft 16½ inches. The thrust blocks are of the ordinary horseshoe type, of white metal, and there are eight rings. The blades of the propeller are of manganese bronze, while the boss is of cast steel.

The condensers are oblong, and support the backs of the cylinders. They are of cast iron, and are fitted with brass tubes, the cooling surface being about 7,000 square feet for each condenser. The condensing water to each condenser is circulated by a large centrifugal pump, driven by an independent engine,

the connection being made between the two condensers. The air pumps are driven by a lever working from the crosshead of the second low-pressure engine in the usual way. There are also fitted two large evaporators to produce the necessary fresh water from the sea water, to make up the feed, and to avoid the use of salt water in the boilers. Two large feed heaters and filtering arrangements are provided. An auxiliary condenser has, in addition, been fitted on board with a separate circulating pump, so that all the auxiliary machinery in the ship is worked separately from the propelling engines.

Boilers.—There are eight single-ended boilers in the ship, constructed of steel and adapted for a working pressure of 300 pounds per square inch. There are four furnaces in each boiler, making thirty-two in all, these being Brown's suspension type. The boilers are fitted with Serve tubes, 3¼ inches in diameter. Messrs. Ellis and Eaves' system of induced draft is fitted to the boilers. The boilers are in two compartments, leading into two funnels, which are 98 feet high from the grate level and elliptical in plan, 13 feet 6 inches by 8 feet 6 inches. The fans are eight in number and 7 feet 6 inches in diameter, and are driven direct by Sturtevant direct-acting engines. These fans, situated directly under the funnels, induce a draft through the furnaces, the air having previously been heated by passing through tubes placed in the way of the waste gases from the furnaces.—"Engineering."

La Lorraine.—Agent Bocande, of the French Line, has received a dispatch from his home office that the line's b'g, new twin-screw steamship *La Lorraine* had arrived at Havre after a successful sea-trial trip. She averaged 22 knots, thus proving herself the fastest French merchantman afloat. If she maintains this average on a voyage she will be about equal to the *Lucania* and *Campania* of the Cunard Line. She measures 12,000 tons and has engines of the triple-expansion type with four cylinders, which were built to develop about 22,000 horsepower. She is 580 feet 8 inches long, of 60 feet 2½ inches beam, and 39 feet 4 inches deep.

S. S. New York.—While bound West on May 22, the Ameri-

can liner *New York* broke her port outboard shaft, and the strut being unable to carry the weight of the tail shaft and propeller, broke and both were lost. The mishap occurred without any warning, and the automatic device with which the engines are fitted instantly brought the gear to mid position and prevented a probable repetition of the mishap to the *Paris* some years ago. Steam was crowded on the starboard engine and the remainder of the voyage was made at about 15 knots rate, the liner reaching New York May 28. It was first intended to send her across for repairs, as the spares were at Southampton, but a careful examination of the starboard engine revealed symptoms of weakness in the crank shaft, and the vessel was sent to Newport News for an overhaul.

S. S. Grosser Kurfurst.—The latest addition to North German Lloyd fleet, the *Grosser Kurfurst* arrived in New York a few days ago. The *Grosser Kurfurst* will be placed in the service between Bremen and New York. She shows a number of improvements as compared with her companion vessels, among which there is especially to be noted one large dining room, first cabin, instead of two smaller dining rooms, as at present in use on the Barbarossa steamers. The dimensions of the *Grosser Kurfurst* are as follows: Length over all, 581 feet 6 inches; beam, 62 feet; depth, 39 feet. The steamer measures 12,200 tons register, has a displacement of 22,000 tons and a capacity of 12,000 tons dead weight. Her motive power consists of two quadruple-expansion engines, developing an average speed of 15 knots across the ocean. Her first cabin will accommodate 250 passengers; her second, 150, and on the two large between decks a large number of steerage passengers can be berthed. The steamer is provided with bilge keels. The cargo space, of which a large part is constructed as a refrigerator for the transportation of perishable goods, is reached by eight large hatches worked with four hydraulic cranes and eleven large winches. The rudder is moved by two heavy steam engines, of which one is held in reserve and used in case of an emergency. These engines are worked from the bridge by hydraulic power. The electric installation consists of 1,000 incandescent lamps, re-

ceiving their electricity from three dynamos. The safety appliances consist in a double bottom running under the whole length of the steamer, and in twelve bulkheads extending to the upper deck. The steamer carries twenty-two large lifeboats, ready for instant use, on the awning deck. The motive power consists of two quadruple-expansion engines of 8,000 indicated horsepower. The cylinders measure 27⅞, 41, 59 and 83½ inches in diameter, and the stroke is 53⅛ inches. The steamer is moved by two four-bladed bronze screws, making ninety revolutions a minute. The steam is generated in five cylindrical double boilers, and two single boilers with natural draft. The boilers are arranged in two groups with two smoke stacks. The heating surface of the boilers is 26,000 square feet, the grate surface, in thirty-six furnaces, is 700 square feet, and the steam pressure is 213 pounds.—"Marine Review."

S. S. Deutschland.—The performance of the *Deutschland* in exceeding on her maiden trip all previous records for the westward passage across the Atlantic suggests that marine engineering has by no means reached the limit of its possibilities. Great as the development of the fast Atlantic steamship has been in the closing years of the nineteenth century, the present indications are that in size, speed, and, above all, in economy of performance, each successive vessel in the opening years of the twentieth century will continue to show the same all-round improvement over its predecessors.

The long looked-for maiden trip has now been run, and a new set of records has been established. The contract for the ship called for a speed of 23 knots, and it seems not too much to believe that this figure as an average speed will be not only reached but even exceeded when the machinery has become adjusted and settled into good working condition.

The *Deutschland* left Plymouth on July 6, at 10·51 P. M., and arrived off Sandy Hook on July 12, at 10·15 A. M. The total distance run was 3,044 knots, and the average hourly speed, 22.42 knots. The daily runs were as follows: 308, 557, 553, 551, 532, 543. There are no records from Plymouth to Sandy Hook, but the distance run is almost the same as that covered on the maiden trip of the *Kaiser Wilhelm der Grosse* from th

Needles to Sandy Hook—3,050 miles, which required 5 days 22 hours 35 minutes, giving an average speed of 21.39 knots.

The *Deutschland* has also beaten the best western run of the *Kaiser Wilhelm der Grosse*, which was made in November last, from Cherbourg to Sandy Hook at an average speed of 22.19 knots.

On the home run eastward the results were still more gratifying, and new records in this direction have also been established. The distance run from Sandy Hook to Plymouth was 3,085 knots, and the time was 5 days 14 hours 6 minutes, giving an average speed of just 23 knots for the trip. The daily runs were as follows: 536, 534, 515, 530, 535 and 435. The previous Sandy Hook-Plymouth record eastward had been held by the *Kaiser Wilhelm der Grosse*, made on her first trip in October, 1897. The time then required was 5 days 15 hours 10 minutes, over a distance of 2,962 knots, giving an average speed of 21.91 knots. The new record betters the time by 1 hour and 4 minutes, over a slightly longer course, making a gain of 1.09 knots average speed for the run.

On the long-distance deep-sea trial, from the mouth of the Oder to the mouth of the Elbe, the mean indicated horsepower was 35,200. On the run from Plymouth to New York, the average indicated horsepower for twenty-four hours was 34,200. The merit of this performance is greatly enhanced by the fact that the total consumption of fuel for the twenty-four hours was 550 tons, which works out at the remarkably low figure of 1.5 pounds per horsepower hour.

ATLANTIC STEAMSHIPS BUILT AND BUILDING.

Name.	Owners.	Length in feet.	Displacement in tons.	Horsepower.	Speed in knots.
Kaiser Wilhelm	N. G. Lloyd	649	20,000	28,000	22½
Oceanic	White Star	704	28,500	28,000	20½
Deutschland	Hamburg-American	686	23,000	34,200	23
Kronprinz	N. G. Lloyd	660	21,500	32,000	23
Kaiser Wilhelm II.	N. G. Lloyd	705	26,000	38,000	23½
Unnamed	White Star	750	32,000	(?)	(?)

That the Atlantic steamship will continue to grow in size and speed is suggested by the three great ships which are now being built, two of them at the Vulcan Yards, Stettin, for the North German Lloyd Company, and one by Harland & Wolff, at Belfast, for the White Star line. The particulars of these vessels are tabulated on the preceding page in comparison with those of recent Atlantic liners.

The North German Lloyd Company will first place in service a vessel, the *Kronprinz*, which in size will come between the *Kaiser Wilhelm* and the *Deutschland*, and in speed will equal the latter. She will be 660 feet long, of 21,500 tons displacement, and is to show a sea speed of 23 knots with 32,000 horsepower. The other vessel, to be known as *Kaiser Wilhelm II*, is to be 705 feet long, of 26,000 tons displacement, with an indicated horsepower of 38,000, and is to maintain a sea speed of 23½ knots. The North German Lloyd Company have not favored the use of superheated forced draft; but if, as is not unlikely, they are moved by the economical steaming of the *Deutschland* to substitute forced for natural draft, we shall expect to see this vessel average 24 knots after she has been a few months in service.—"Scientific American."

S. S. Californian.—This large steamship for the new American-Hawaiian line was launched at the Union Iron Works, San Francisco, May 12. She is of these dimensions: Length, 430 feet; beam, 50 feet; depth, at center, 34 feet 6 inches; draught, 26 feet. She is constructed in accordance with Lloyd's three-deck rules. Triple-expansion engines will be fitted with cylinders, 27 inches, 45½ inches and 76 inches, by 48 inches stroke, capable of developing 2,500 I.H.P. Steam will be supplied by four single-ended Scotch boilers. With 8,250 tons dead weight a mean speed of about 10 knots will be maintained at sea. The bunker capacity is 1,500 tons. This is the largest vessel ever built on the Pacific coast. Two similar ships are now under construction at the Roach yards, Chester, Pa. The *Californian* was launched by moonlight between 10 and 11 P. M. Miss Edith Chesebrough, daughter of the vice-president of the owning corporation, broke the customary bottle of wine.

ASSOCIATION NOTES.

In accordance with the By-Laws of this Society, a meeting will be held on Tuesday, October 2, 1900, for the purpose of nominating officers for the coming year. As these nominations are to be submitted in writing, any member desiring to propose a name or names for the offices should send the same in before the above date, addressed to the Society, and marking on the outside of the envelope the words "Nominations—to be opened at October 2 meeting."

It has been suggested that the interests of the JOURNAL might be best served by the election of a suitable retired engineer officer as Secretary-Treasurer, owing to the fact that those on active duty in Washington are prevented by press of official work from devoting the time properly required for extending the influence of the periodical and advancing its importance in a ratio commensurate with the character of its advantages. The editorial work necessarily devolves upon the Secretary-Treasurer, in addition to the extensive correspondence attached to the circulation and advertising, so that this suggestion is here given in order to lead to a possible improvement in the conductorship of the work.

Nominations are to be made for President, Secretary-Treasurer and three names for Council. Also any suggestions which any member may desire to submit to the Society for consideration at the meeting should be promptly forwarded, if the member is unable to be present in person.

While the character of the JOURNAL is such as to make it to a great degree eclectic, the idea being to collate and condense important data as gathered from culling a large range of exchanges, it is also highly desirable that it should be the pro-

mulgator of as much original engineering research and experiment as possible, as well as circulating such food for thought as is so often contained in a clear description of discovered unusual conditions of machinery, either in operation of the same or as results of casualty or distress. The JOURNAL should be the means of such exchange of interesting and instructive experiences, and this can readily be accomplished if our members will more fully realize the importance of this and will forward to the JOURNAL the results of their investigation or observation in matters of professional engineering.

It should be noted that sketches or drawings sent as illustrations must, to be useful, be absolutely clear and carefully made, and also that photographs are best reproduced if made on solio or other glossy-surface paper.

ERRATA.

Page 895, second line, should read "*John W. Gates*, owned by the American Steamship Co."

Page 990, third line from bottom: Pitch ÷ diameter should read 1.13 instead of 113.